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Heating · Piping and Air Conditioning

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APRIL, 1937

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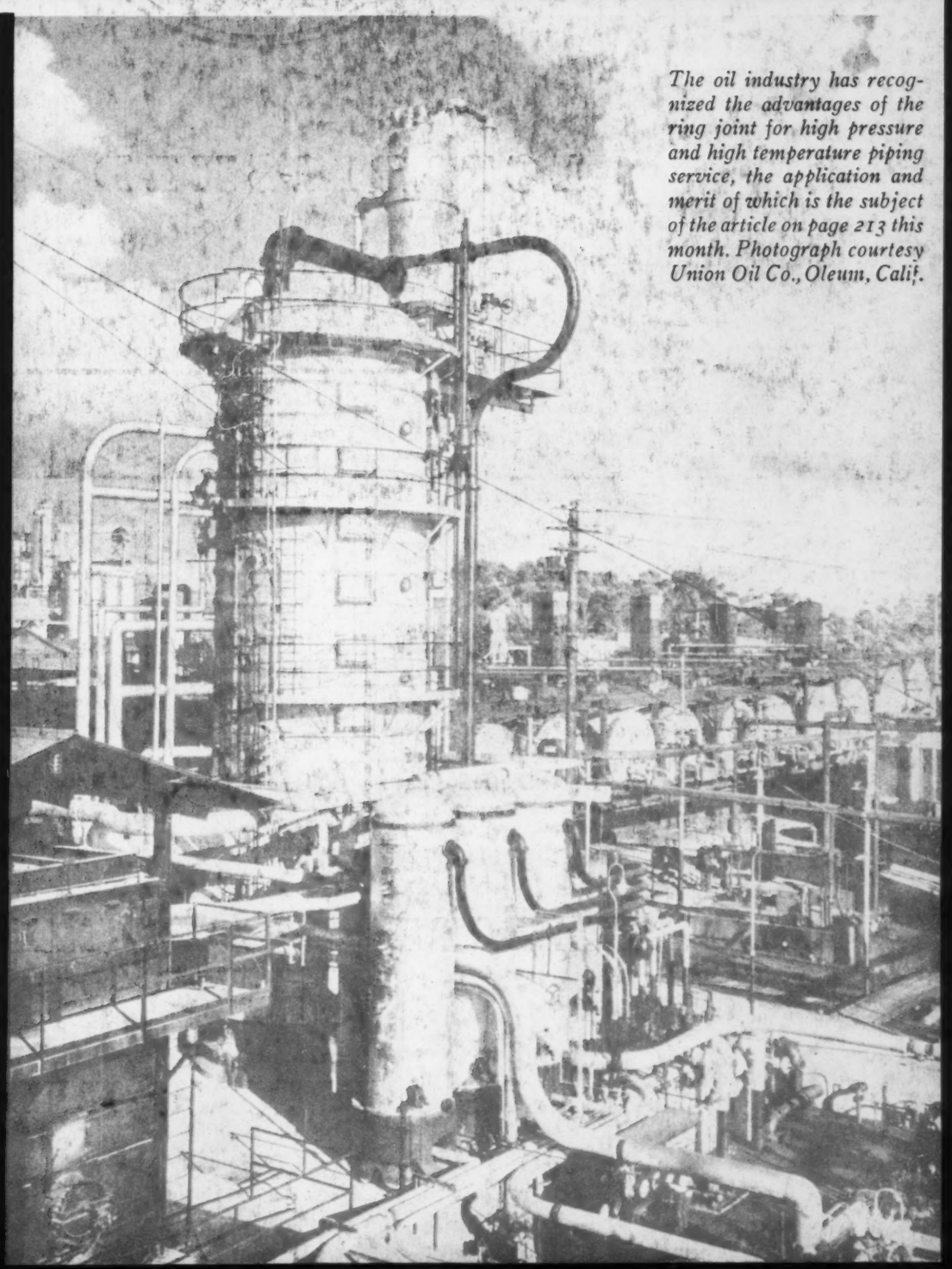

The oil industry has recognized the advantages of the ring joint for high pressure and high temperature piping service, the application and merit of which is the subject of the article on page 213 this month. Photograph courtesy Union Oil Co., Oleum, Calif.

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The Editor's Page

■ **Merged**—To HEATING, PIPING AND AIR CONDITIONING'S "mast-head" this month is added the name of a publication well known in the air conditioning industry since 1925 . . . "The Aerologist." To HEATING, PIPING AND AIR CONDITIONING'S Board of Consulting and Contributing Editors this month is added the name of Dr. E. Vernon Hill, editor and publisher of "The Aerologist" since its inception. All of which means that Keeney Publishing Company has purchased "The Aerologist" and merged it with HEATING, PIPING AND AIR CONDITIONING. In the editorial pages this month is Dr. Hill's first article, a most interesting suggestion for meeting the supply and disposal problem created by condensing water requirements of air conditioning installations.

■ **A district heating competition** is being sponsored by the National District Heating Association for newer members, those who have—or will have—joined the Association between January 1, 1934, and May 1, 1937. Deadline for the papers submitted, which must be on some phase of the subject of generation, distribution and sale of steam or any of its attendant ramifications, is May 1, and winners will be announced at the annual convention in Detroit, May 25-28. Entries are to be mailed to R. M. Nee, 39 Boylston St., Boston, Mass. . . . Relative importance of the factors for selecting coal has been evaluated in a series of charts to guide the consumer in buying coal, and the producer in reaching markets for which his product is best suited; seven different use classifications are included: stationary steam generation, hand fired; stationary steam generation, stoker fired; stationary steam generation, pulverized coal; coke and gas making; ceramic products; miscellaneous; domestic, bunker and cargo. . . . Titled "Factors Recommended for Consideration in the Selection of Coal," copies of the publication are available from the American Standards Association, 29 W. 39th St., New York City, at $1.00.

■ **A business executive's Btu's** suffice to bring 5.8 gallons of water to boiling, according to the Temperature Research Foundation of Nash-Kelvinator Corp.; the average executive dissipates 8789 Btu in a normal 24 hour day, and his secretary will expend more than that—9469 Btu being the figure given.

■ **Brewster S. Beach,** member of our Board of Consulting and Contributing Editors, and William H. Baldwin announced last month their association as consultants in public relations and publicity, with offices in New York City. . . . William Goodman, Consulting and Contributing Editor, will speak at the air conditioning session of the conference on air conditioning and automatic heating at the University of Wisconsin, April 21-23. . . . Willis H. Carrier is to address an evening session on the future of air conditioning, and Prof. G. L. Larson, chairman of the University's mechanical engineering department, will speak on the human power plant. Physiological factors in air conditioning, stokers, control, hot water service, gas firing, air distribution, air filters, oil firing, and insulation are among the subjects on the program. . . . A uniform external temperature was cited by Mr. Carrier as the greatest essential in the prevention of the common cold at the National Oil Burner and Air Conditioning Exposition last month. . . . Installed cost of air conditioning equipment sold by members of the Air Conditioning Manufacturers' Association for the first two months this year represented an increase of 165 per cent over the first two months of 1936, according to William B. Henderson, executive vice-president.

■ **Air conditioned bedrooms** at the Hotel Statler, St. Louis, proved so popular last year that contracts have been let for equipping the remaining rooms (one-half of the rooms were conditioned last summer) which will give the Statler more conditioned rooms than any other hotel, according to Frank A. McKowne, president of Hotels Statler Co., Inc. . . . Six thousand horsepower will be required to operate the refrigerating plant for air conditioning purposes in the new Capitol power plant building, Washington, D. C., which will supply 11,500,000 gallons of chilled water daily through approximately a mile of underground piping, according to York Ice Machinery Corp., who has received the contract for this refrigerating plant. . . . A $556,928 contract to complete the air conditioning system in the U. S. Capitol building, home of the Senate and House of Representatives, has been awarded Carrier Corp., it was announced last month. The new equipment will extend the system installed by Carrier in 1928 that operated only in the chambers of the Senate and the House. . . . First air conditioning contracts for educational institutions in Chicago were closed recently, one for the new administration building at the University of Chicago, and one for DePaul University's downtown school. . . . Cows in hot and humid Singapore are living in a barn cooled by air conditioning in a move by Malayan dairymen to reduce adverse effects of the climate on milk production.

■ **Conferences** on heating and air conditioning will be held at Iowa State College, Ames, Iowa, April 7-9, and at the University of Wisconsin, Madison, April 21-23, under the auspices of the mechanical engineering departments of the respective institutions. . . . The American Foundrymen's Association will meet in Milwaukee, May 3-7. . . . An exposition of power and mechanical engineering will be held in Chicago, October 4-9, at the International Amphitheatre, and under the management of the International Exposition Co., manager of the biennial National Exposition of Power and Mechanical Engineering in New York City; the next New York exposition is scheduled for December, 1938.

■ **The outlook** is for continued improvement in building activity, said Clarence M. Woolley, chairman of American Radiator and Standard Sanitary Corp., in reporting to stockholders largest profits since 1929 during 1936.

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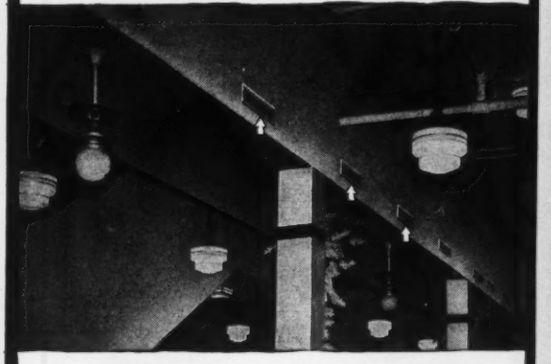

...THE SOLUTION... working with Architect Solomon Kaplan, Carrier engineers selected the particular type of equipment best suited. Special ductwork was designed, to avoid the maze of pipes made necessary by each building's individual heating plant—and a method devised to control temperature in each building.

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Heating · Piping and Air Conditioning

with which is merged the "Aerologist"

Design, Installation
Operation, Maintenance
in Industry and Large Buildings

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CONTENTS *for* APRIL, 1937

Volume 9 Number 4

The Ring Joint—Its Relative Merit and Application, by E. C. Petrie 213

How to Avoid Trouble on Cooling Jobs With Evaporators Above Condensers, by William Goodman 221

Superheated Water System Heats New Swiss Buildings, by R. Ruegg 224

University of Illinois Installs New Experimental Conditioning Equipment, by William H. Severns and M. K. Fahnestock 227

How to Operate Centrifugal Pumps 231

The Problem of Condenser Water for Large Air Conditioning Plants, by E. Vernon Hill 233

Performance of Heaters and Coolers Under Changed Operating Conditions, by Bayard P. Fonda 237

Electricity Used by Air Conditioning, by William P. Rock 239

Accessibility of Service Piping Features Aircraft Assembly Plant 242

The Strength and Design of Covers and Flanges for Pressure Vessels and Piping, by T. McLean Jasper, H. Gregersen and A. M. Zoellner 243

Controls, by C. L. Ringquist 245

Heating, Piping and Air Conditioning Summary Sheets—Properties of Pipe 247

Equipment Developments (back section) 132

Question on Heating Bunkhouse Answered, by Samuel R. Lewis (back section) 148

Recent Trade Literature (back section) 152

Index to Advertisers (back section) 160

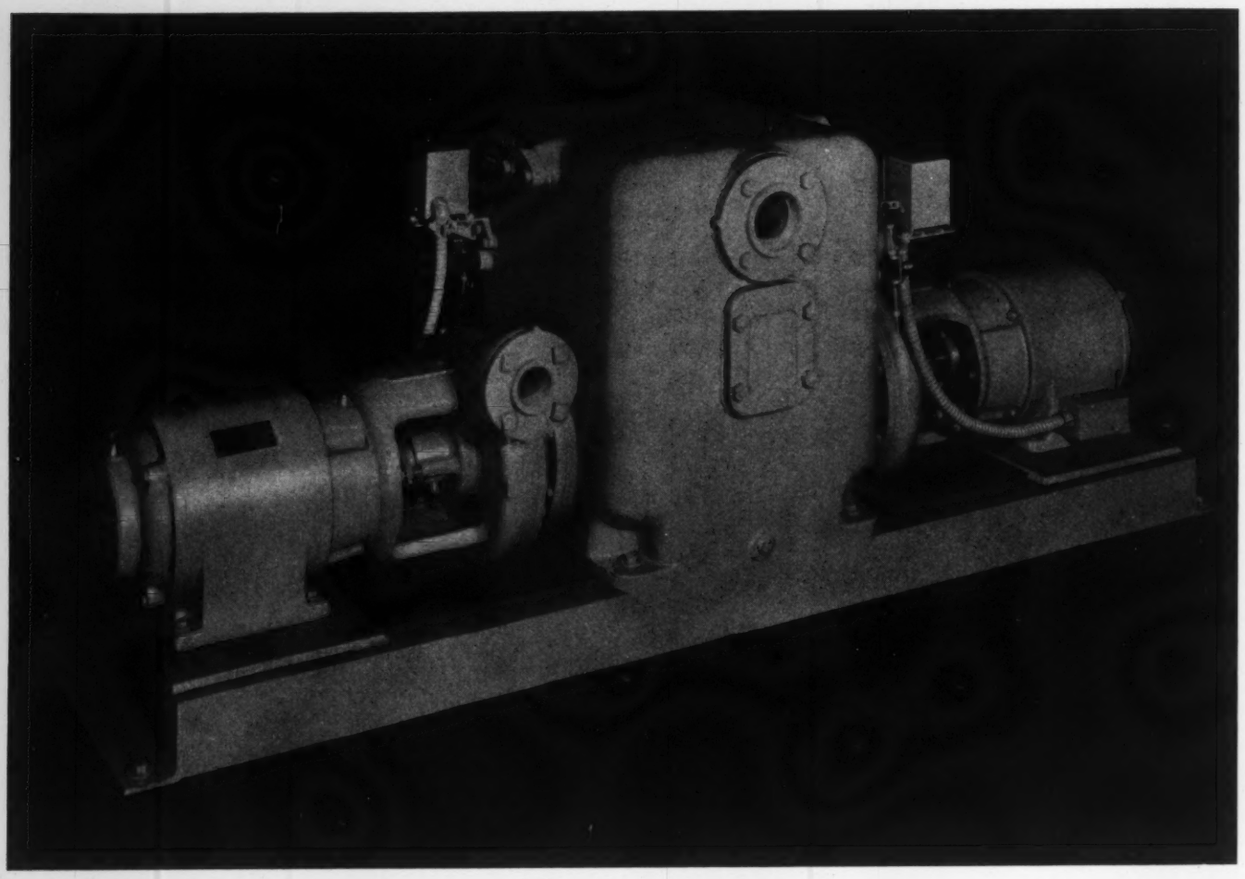
Journal Section of the American Society of Heating and Ventilating Engineers

The Cooling and Heating Rates of a Room With Different Types of Steam Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick 251

Summer and Winter Air Conditioning by Low Pressure Steam, by G. F. Zellhoefer 265

Bulletin Section 271

MAXIMUM EFFICIENCY FOR STEAM HEATING SYSTEMS ON HEATING, DRYING AND PROCESS WORK.



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Heating·Piping and Air Conditioning

Vol. 9, No. 4 • *April, 1937*

THE RING JOINT

Its Relative Merit and Application

By E. C. Petrie*

The application of the ring joint to high pressure and high temperature piping service has been a step in the right direction to minimize leakage at flanged joints, the author believes. The data he presents indicate the advantage of this type of joint over the conventional flat gasket joint. . . . Further tests are being conducted to determine the effect of high temperatures on the pressure retaining ability of the ring joint, and therefore the information given should be considered more or less as a progress report and not as a presentation of definite conclusions. . . . A committee of the *American Petroleum Institute* has recently proposed dimensions of rings for a joint standard, which are tabulated here.

USE of steel valves and fittings for high pressures and temperatures has made the problem of maintaining tight flanged joints a serious one. High pressures have required the use of heavy flanges and large size bolts, while high temperatures have necessitated a search for materials that will exhibit the greatest resistance to plastic deformation, or creep.

The size of the bolts required to produce a total bolt load necessary to maintain a tight joint is largely dependent upon the type and dimensions of the gasket used. For example, it has been shown by many investigations that some types of joints will resist a higher internal pressure before joint leakage occurs under the same total bolt load than flat gasket types of joint, such as the raised male face to male face type. Therefore, it would seem reasonable to assume that the size of bolt for the flange having the better joint could be smaller for the same pressure rating than the size of bolt for the flat gasket type. Practically, however, as flanges can be supplied with any type of facing, the bolts are made large enough to maintain a tight joint under average operating conditions with the flat face gasket. This leads to the conclusion that if the gasket joint *performance factor*, which may be described as the total load due to internal pressure acting on an area bounded by the outside bearing circumference of the gasket face divided by the total bolt load applied, is greater than that offered by a flat gasket, the safety factor based on the ratio of the leakage pressure to the working pressure will be increased proportionately.

During the past decade a type of joint has developed which has proved itself to be superior to the flat type both in actual practice and in laboratory tests. This joint, known commercially as the ring joint, comprises a solid ring which is retained between tapered straight sided grooves, having a pitch diameter equal approximately to the outside diameter of the male face of an *A.S.A.* steel flange as shown in Fig. 1. It has been used extensively, especially in oil refinery and production service where it has been subjected to very high temperatures and pressures with satisfactory results. The ability of the ring joint to withstand pressures greatly in excess of those which can be retained by the regular flat gasket type, such as the male and female, tongue and groove, etc., has promoted its use where high pressures are prevalent and has also aroused the interest of research investigators to such an extent that many tests have been conducted to compare the pressure retaining ability of the ring joint with that of a flat gasket type of joint. As a general rule, flat gasket types of joints yield performance factors of 0.50 to 0.75, whereas the performance factors of ring joints are rarely found to be below 1. The oil industry has recognized the ring joint and is at the present time developing an *A.P.I.* standard to cover the dimensions of ring joints for both refinery and production services.

It is our purpose here to compare the ring joint with the flat gasket type. The following subjects are discussed:

A. General characteristics of a ring joint flanged connection.
B. Effective pressure area of ring joints.
C. Results of comparative tests of ring and flat gasket joints.
D. Interpretation of test results.
   a. Ring joints vs. flat gasket joints.
   b. Effect on pressure retaining ability of ring joints.
      1. Flange thickness.
      2. Width of ring.
      3. Shape of ring.
E. Method of determining width of ring section for ring joints.
F. Application of the ring joint.
   a. Proposed dimensions for ring joints for an *A.P.I.* standard.

General Characteristics of Ring Joint Connection

In order to analyze the action of flanged joints under combined bolting and pressure loads, let us consider for a comparative example two flanged connections, one

*Research Testing Laboratories, Crane Co., Chicago, Ill.

having a male to male joint with a flat gasket, the other a ring joint, as shown in Figs. *2a* and *2b*.

It is well known that bolts elongate in direct proportion to their respective cross sectional areas when stressed by the application of a turning moment to the nuts. For example, if the bolt material is alloy steel, having a modulus of elasticity of approximately 29,000,000 at room temperature and if the bolt has a uniform cross sectional area of 1 sq in. throughout its entire length (that is, either it is threaded for its entire length or the distance between the threaded portions is machined down to the root diameter of the thread) it will elongate 0.001 in. per in. of length under a tensile pull of 29,000 lb. Tables 1 and 2 give the empirical relationship between turning effort applied to well lubricated nuts and the stress and total compression produced by bolts having 8 pitch and coarse threads.[1]

As the bolts elongate under stress, the flanges have a tendency to deflect and the gasket or ring will be compressed, more so at the outer circumference than at the inner. This is true in almost all practical installations as the flanges are not as a general rule rigid enough to eliminate all flange deflection. Examination of many gaskets, both of the flat and ring type, has shown that the intensity of the bearing at the outer circumference is more pronounced than at the inner.

When internal pressure is applied to both joints, the pressure will act in a direction which will tend to separate the flanges in each case. Due to the fact that the gasket has elastic properties, potential energy is stored up in it when the bolt load is applied. It has been indicated by many tests that the release of this energy by the application of internal pressure causes a slight increase in bolt stress before the total bolt load is equaled by the total pressure load. This increase in stress may be measured by noting the additional elongation of the bolts under pressure. As the bolts elongate, the distance between the nut faces increases, consequently decreasing the effective gasket compression.

In the flat gasket type of joint this premature release of gasket compression generally causes the joint to leak before the total pressure load reaches or even approaches the total bolt load applied. This is particularly so for wide gaskets where the unit compressive load is relatively small with respect to that which would be obtained with a narrow gasket.

This same premature bolt elongation takes place when internal pressure is applied to the ring joint type of flanged connection, but there are other forces which act on a ring gasket that are not present in a flat gasket type of joint. When internal pressure is applied to a circular ring, the ring expands diametrically and causes a force to be applied to the outside contact bearing of the groove in which it is located. Therefore, even though the flanges separate slightly due to additional bolt elongation, the diametrical force applied by the expanding ring will tend to maintain a tight joint. Other conditions which may affect the pressure retaining ability of ring joints are (1) the wedging effect of the ring in the groove; (2) the deflection of the flanges; (3) the tendency of the ring and groove to remain in contact for

[1] "Flanged Piping Joints—Methods for Applying and Determining Bolt Stresses," by E. C. Petrie. HEATING, PIPING AND AIR CONDITIONING, June, 1936, pp. 303-309.

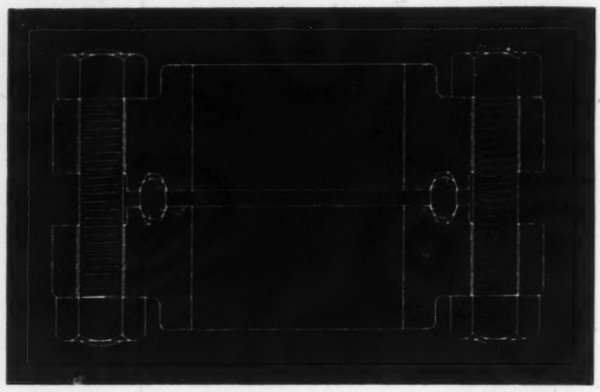


Fig. 1—Ring type flanged joint

Table 1—Torque-Stress Relationship for Alloy Steel Bolts Load in Pounds on Bolts When Torque Loads Are Applied

Nom. Dia. of Bolt, In.	No. of Th'ds per Inch	Dia. at Root of Thrd.	Area at Root of Thrd.	Stress: 30,000 Lb per Sq In. Torque Ft-Lb	30,000 Lb per Sq In. Compression Lb	45,000 Lb per Sq In. Torque Ft-Lb	45,000 Lb per Sq In. Compression Lb	60,000 Lb per Sq In. Torque Ft-Lb	60,000 Lb per Sq In. Compression Lb
1/4	20	0.185	0.027	4	810	6	1215	8	1620
5/16	18	0.240	0.045	8	1350	12	2025	16	2700
3/8	16	0.294	0.068	12	2040	18	3060	24	4080
7/16	14	0.345	0.093	20	2790	30	4185	40	5580
1/2	13	0.400	0.126	30	3780	45	5670	60	7560
9/16	12	0.454	0.162	45	4860	68	7290	90	9720
5/8	11	0.507	0.202	60	6060	90	9090	120	12120
3/4	10	0.620	0.302	100	9060	150	13590	200	18120
7/8	9	0.731	0.419	160	12570	240	18855	320	25140
1	8	0.838	0.551	245	16530	368	24795	490	33060
1 1/8	8	0.963	0.728	355	21840	533	32760	710	43680
1 1/4	8	1.088	0.929	500	27870	750	41805	1000	55740
1 3/8	8	1.213	1.155	680	34650	1020	51975	1360	69300
1 1/2	8	1.338	1.405	800	42150	1200	63225	1600	84300
1 5/8	8	1.463	1.680	1100	50400	1650	75600	2200	100800
1 3/4	8	1.588	1.980	1500	59400	2250	89100	3000	118800
1 7/8	8	1.713	2.304	2000	69120	3000	103680	4000	138240
2	8	1.838	2.652	2200	79560	3300	119340	4400	159120
2 1/4	8	2.088	3.423	3180	102690	4770	154035	6360	205380
2 1/2	8	2.338	4.292	4400	128760	6600	193140	8800	257520
2 3/4	8	2.588	5.259	5920	157770	8880	236655	11840	315540
3	8	2.838	6.324	7720	189720	11580	284580	15440	379440

Table 2—Torque-Stress Relationship for Coarse Thread Bolts Load in Pounds on Bolts When Torque Loads Are Applied

Nom. Dia. of Bolt, In.	No. of Th'ds per Inch	Dia. at Root of Thrd.	Area at Root of Thrd.	Stress: 7,500 Lb per Sq In. Torque Ft-Lb	7,500 Lb per Sq In. Compression Lb	15,000 Lb per Sq In. Torque Ft-Lb	15,000 Lb per Sq In. Compression Lb	30,000 Lb per Sq In. Torque Ft-Lb	30,000 Lb per Sq In. Compression Lb
1/4	20	0.185	0.027	1	203	2	405	4	810
5/16	18	0.240	0.045	2	388	4	675	8	1350
3/8	16	0.294	0.068	3	510	6	1020	12	2040
7/16	14	0.345	0.093	5	698	10	1395	20	2790
1/2	13	0.400	0.126	8	945	15	1890	30	3780
9/16	12	0.454	0.162	12	1215	23	2430	45	4860
5/8	11	0.507	0.202	15	1515	30	3030	60	6060
3/4	10	0.620	0.302	25	2265	50	4530	100	9060
7/8	9	0.731	0.419	40	3143	80	6285	160	12570
1	8	0.838	0.551	62	4133	123	8265	245	16530
1 1/8	7	0.939	0.693	98	5190	195	10380	390	20760
1 1/4	7	1.064	0.890	137	6675	273	13350	545	26700
1 3/8	6	1.158	1.054	183	7905	365	15810	730	31620
1 1/2	6	1.283	1.294	219	9705	437	19410	875	38820
1 5/8	5 1/2	1.389	1.515	300	11363	600	22725	1200	45450
1 3/4	5	1.490	1.744	390	13080	775	26160	1550	52320
1 7/8	5	1.615	2.049	525	15368	1050	30735	2100	61470
2	4 1/2	1.711	2.300	563	17250	1125	34500	2250	69000

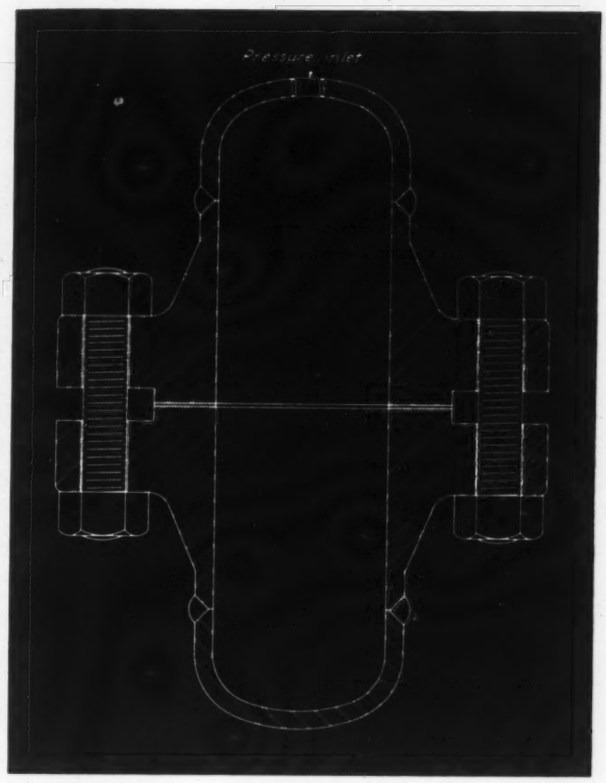
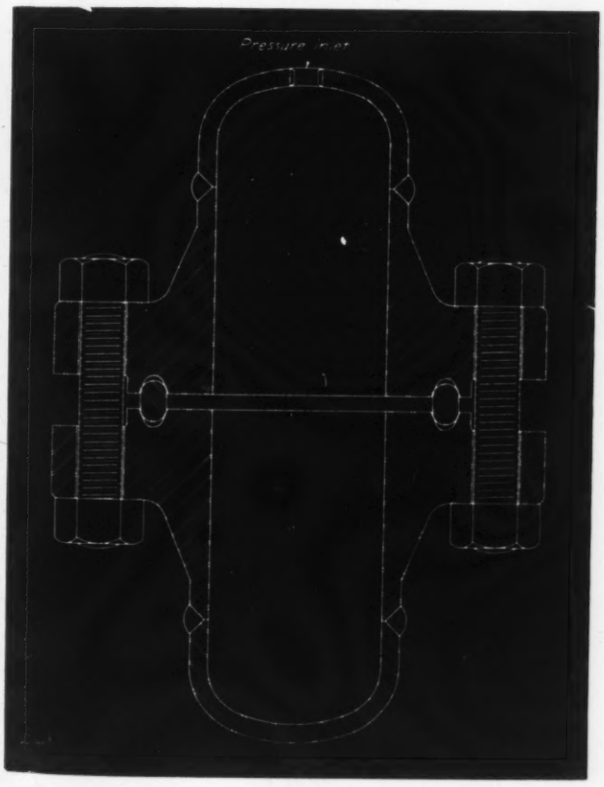


Fig. 2*a*—Male to male flat gasket joint

Fig. 2*b*—Ring joint



a longer period of time than the faces of a flat gasket type of joint due to the tapered sides of the grooves.

Effective Pressure Area of Ring Joints

In determining the total pressure load for flanged joints, the assumption has been made that at the instant the joint leaks the total area upon which the internal pressure is acting is that bounded by the outside circumference of the contact surface. This assumption is apparently true on flat gasket joints, but in order to prove its validity for a ring joint connection the following test was conducted.

The flanges of a 6 in., 400 lb *A.S.A.* cast steel tee were machined for ring joints, using rings having widths and pitch diameters of $\frac{7}{16}$ in. and $8\frac{5}{16}$ in. respectively. The machined rings were made of soft steel having a Brinell hardness of approximately 90. Blind flanges, machined with the same grooves, were bolted to the tee flanges with twelve $\frac{7}{8}$ in. full threaded alloy steel bolt studs in each joint. Tapped holes for $\frac{1}{8}$ in. pipe were drilled from the backs of the blind flanges into the tops of the machined grooves in order to determine the instant at which leakage would occur at the inner circumferences of the rings.

The nuts of the bolt studs in each joint were pulled up with a torque of 160 ft-lb to obtain a bolt stress of 30,000 lb per sq in. The deflection of the flanges and the depression of the soft steel rings in the grooves were noted under this bolt load. It was found that while the ring settled into the grooves approximately 0.030 in., the flanges did not bend perceptibly.

Internal hydrostatic pressure was then applied to the tee and it was noted that leakage occurred at the bearing surfaces at the inner circumferences of the rings at approximately 3200 lb. The small drilled holes were than plugged and at about 3900 lb pressure the ring joints leaked at the outer bearing surfaces.

Table 3 gives a comparison of the total load exerted by the internal pressure and the initially applied bolt load.

The results of this test indicate that leakage takes place at the inner bearing surfaces of ring joints before leakage takes place around the outer circumference of the ring. It will also be noted by referring to Table 3 that the relationship between the bolt load initially applied and the pressure load to cause leakage at the outer circumference may be expressed as 234,390 ÷ 150,840 or a performance factor of 1.56.

Results of Comparative Tests of Ring and Flat Gasket Joints

Table 4 gives the results of many tests conducted on both ring joints and flat gasket joints and illustrates the relative pressure retaining ability of both types. In all cases the total pressure load is based upon the area bounded by the outer bearing circumference of the gasket.

Table 3

Total Bolt Load, Pounds (Twelve ⅞ inch Bolts 30,000 Lb per Sq In. Stress)	Force Exerted Against Flange by 3200 Lb[a]	Force Exerted Against Flange by 3900 Lb[b]
150,840	156,800	234,390

[a]Based on area bounded by inner bearing circumference of ring = 49 sq in. × 3200 lb.
[b]Based on area bounded by outer bearing circumference of ring = 60.1 sq in. × 3900 lb.

Interpretation of Test Results

Flat Gasket Joints vs. Ring Joints

Table 4 indicates that in almost every case the ring joint application is superior to the flat gasket type of joint; see comparative tests Nos. 1, 2, 3, and 4. The average ratio of total leakage pressure load based on outside circumference of contact surface and the applied bolt load is given in Table 5 for all of the joints listed in Table 4.

Effect of Flange Thickness on Pressure Retaining Ability of Ring Joints

With the thought that flange deflection might have

Table 4—Test Data

Test Number	Size of Flange Nominal & O. D.	No. and Size of Bolts	Gasket				Applied Bolt Stress, Lb per Sq In.	Applied Total Bolt Load Lb B	Leakage Press., Lb per Sq In.	Total Internal Pressure Lb A	Performance Factor A/B
			Type of Joint and Gasket Material	O. D.	I. D.	O. D. Area Sq In.					
1	4 x 14 5/8	12—1 1/4	Male and Female. Akro Metal	9 1/16	7 3/8	64.4	40500 48000 66000	452000 536000 737000	5000 6500 10000	322000 418000 644000	0.712 0.780 0.874
	4 x 14 5/8	12—1 1/4	Oval Ring Joint	9 29/32	9 1/32	77	33600	312000	14000	1077000	3.45
2	6 x 12 1/2	16—5/8	Male and Female. Akro Metal	8 7/8	7 3/4	61.9	35000 70000 105000 120000	113000 226000 339000 385000	1600 2200 3300 4000	99000 136000 204500 247500	0.877 0.602 0.604 0.643
	6 x 12 1/2	16—5/8	Oval Ring Joint	8 31/32	8 3/32	63	70000 105000 120000	226000 339000 385000	4300 5000 5600	271000 315000 353000	1.20 0.93 0.917
3 (Part 1)	12 x 25 1/2	16a—1 7/8	Narrow T & G. Cranite (Asbestos)	14 1/4	13 1/2	159.5	21400 28500	686000 915000	3900 5000	622000 797000	0.908 0.872
	12 x 25 1/2	16a—1 7/8	Wide T & G. Cranite	15	13 1/2	176.7	35700	1144000	6000	1060000	0.927
	12 x 25 1/2	16a—1 7/8	Narrow T & G. Akro Metal	14 1/4	13 1/2	159.5	28500 46800	915000 1500000	5600 6200	893000 990000	0.976 0.660
	12 x 25 1/2	16a—1 7/8	Narrow T&G. Copper	14 1/4	13 1/2	159.5	46800	1500000	6200	990000	0.660
	12 x 25 1/2	16a—1 7/8	Wide T&G. Copper	15	13 1/2	176.7	46800	1500000	5800	925000	0.617
	12 x 25 1/2	16a—1 7/8	Oval Ring Joint	15 5/8	14 3/8	191.7	21400 28500 35700	686000 915000 1144000	4200 6100 7000	806000 1170000 1340000	1.175 1.28 1.17
3 (Part 2)	12 x 25 1/2 (Flg. Thick. 4 1/8)	16a—1 7/8	Oval Ring Joint	(b) 15 5/8	14 3/8	191.7	21400 28500 35700	686000 915000 1144000	4200 6100 7000	806000 1170000 1340000	1.175 1.28 1.17
	12 x 25 1/2 (Flg. Thick. 3 11/16)	16a—1 7/8	Oval Ring Joint	(b) 15 5/8	14 3/8	191.7	21400 28500 35700	686000 915000 1144000	4100 5600 6300	785000 1070000 1205000	1.145 1.17 1.05
	12 x 25 1/2 (Flg. Thick. 3 1/4)	16a—1 7/8	Oval Ring Joint	(b) 15 5/8	14 3/8	191.7	21400 28500	686000 915000	3630 4600	695000 880000	1.015 0.962
	12 x 25 1/2 (Flg. Thick. 3 1/8)	16a—1 7/8	Oval Ring Joint	(b) 15 5/8	14 3/8	191.7	14100 21400	452000 686000	2930 3600	561000 690000	1.24 1.01
4	20 x 38 3/4	16—3	Sarlun Jt. Not Sealed	26 1/16	15	533.5	32100	3248000	4600	2455000	0.756
	20 x 38 3/4	16—3	Oval Ring Joint	23 7/8	22 1/8	447.7	15000 26400	1517760 2671258	5300 7000	2370000 3130000	1.56 1.17
	20 x 38 3/4	16—3	Octagonal Ring Joint	24	22	452	13980 20700	1414550 2095000	5000 7000	2260000 3165000	1.60 1.51
5	12 x 25 1/2	16—1 7/8	Oval Ring Joint	15 5/8	14 3/8	191.7	13200 26400 39600	487840 975680 1463520	4100 7200 8200	786000 1380000 1570000	1.61 1.42 1.08
	12 x 25 1/2	16—1 7/8	Oval Ring Joint	16 1/4	13 3/4	207.4	13200 26400 39600	487840 975680 1463520	3550 5100 8950	736000 1060000 1855000	1.51 1.08 1.27
6	6 x 12 1/2	12—3/4	Oval Ring Joint	8 3/4	7 7/8	60.1	30000 60000 105000	108720 217440 380520	1800 4050 6500	108300 244000 391000	1.00 1.12 1.03
	6 x 12 1/2	12—7/8	Oval Ring Joint	8 3/4	7 7/8	60.1	30000 60000 105000	150840 301680 527940	4050 6000 8650	244000 361000 520000	1.62 1.20 0.99
	6 x 14	12—1	Oval Ring Joint	8 3/4	7 7/8	60.1	30000 60000	198360 396720	4150 7850	250000 472000	1.26 1.19
	6 x 15	12—1 1/8	Oval Ring Joint	8 3/4	7 7/8	60.1	36700	318500	7500	451000	1.42
	6 x 15 1/2	12—1 3/8	Oval Ring Joint	8 3/4	7 7/8	60.1	26400	364800	9500	571000	1.57
7	4 x 10	8—3/4	Oval Ring Joint	6 5/16	5 5/16	31	27000 54000 81000 94500	65200 130400 195600 228000	3500 5700 6900 7400	108400 176500 213600 229000	1.66 1.35 1.09 1.00
8	6 x 15 1/2	12—1 3/8	Octagonal Ring Joint	8 13/16	7 13/16	61.0	15000 17500 30000 45000	207600 242200 415200 622800	5300 6800 12000 16250	323000 414000 731000 990000	1.55 1.71 1.76 1.59
	12 x 25 1/2	16—1 7/8	Octagonal Ring Joint	15 7/8	14 1/8	197.9	15000 30000 45000	552960 1105920 1658880	3900 7800 10000	771000 1542000 1979000	1.40 1.40 1.19

a Bolt studs used were made with 5 threads per inch.

b New rings were used for every flange thickness. Grooves were remachined.

Table 5

	Flat Gasket Type	Ring Joint
Performance Factor.....	0.75	1.25

some bearing on the pressure retaining ability of ring joints, several 12 in. joints were tested with varying thickness of flanges.

Part 2 of Test No. 3 indicates that better results were obtained with the thicker flanges than with the thinner. The data obtained from this test do not indicate that any advantage is gained by abnormal flange deflection; however, the results of a single test are not conclusive as it may be that in another the dishing of the flange may cause a biting action on the outer circumference of the ring which may increase the pressure at which leakage will occur.

Effect of Width of Ring on Pressure Retaining Ability of Ring Joints

Tests Nos. 4 and 5 indicate that rings having a width greater than $\frac{7}{16}$ in. will maintain pressures at least equal to that held by a $\frac{7}{16}$ in. ring.

In order to prove definitely that a heavy ring is equal to a lighter ring with reference to pressure retaining ability the following test was conducted:

Two oval section ring gaskets were made up, both having a pitch diameter of 15 in. and widths of $\frac{7}{16}$ in. and $\frac{7}{8}$ in. respectively. A 12 in. *A.S.A.* 1350 lb spool was machined with a groove for the light section ring in one flange and a groove for the heavy section ring in the other. Two 12 in. 1350 lb forged steel blind flanges were machined with grooves to fit the same rings. Sixteen $1\frac{7}{8}$ in. full threaded alloy steel bolt studs, lubricated with graphite and oil, were used in the joints.

The fit of the ring gaskets in their respective grooves was checked before the tests were started and both rings indicated a uniform bearing on both inner and outer circumferences.

In conducting the tests, the light and heavy section rings were assembled in their respective flanges and all of the bolt studs were pulled up lightly but firmly to set the flanges. The bolts were then pulled up by one man using a 3 ft lever. This was assumed as being the minimum load which would be applied in practice.

The torque applied was measured and was found to be approximately 500 ft-lb, which created a stress of 7500 lb per sq in. in each bolt stud.

After both joints had been pulled up in this manner, the entire assembly was subjected to internal hydrostatic pressure until the joints leaked. This same procedure was followed with bolt stresses of 15,000, 30,000, 40,000, and 45,000 lb per sq in., hydrostatic pressure tests being conducted after each increase in bolt stress.

Table 6—Hydrostatic Pressure Tests

Applied Bolt Stress, Lb per Sq In.	Total Bolt Load *B*	Light Section Ring $\frac{7}{16}$ In.			Heavy Section Ring $\frac{7}{8}$ In.		
		Leakage Press. Lb per Sq In.	Total[a] Press. Load *A*	Performance Factor *A/B*	Leakage Press. Lb per Sq In.	Total[a] Press. Load *C*	Performance Factor *C/B*
7500	276480	2000	374400	1.35	2300	455170	1.64
15000	552960	4200	786240	1.42	4100	811390	1.47
30000	1105920	8000	1497600	1.35	8000	1583200	1.43
40000	1474560	8500	1591200	1.08	9000	1781100	1.21
45000	1658880	9000	1684800	1.01	11000	2176900	1.31

[a]Total pressure load is the product of the leakage pressure and the area bounded by the outer circumference of the ring.

The test results in Table 6 indicate that the thickness of the ring within reasonable limits does not affect the pressure retaining ability of a ring joint. They also show that it is possible to make ring joints with relatively heavy section rings tight under the minimum bolt load which will probably be applied in service. Conversely, however, some tests have indicated that if the ring is too light for the flange bolting and bottoms in the groove, the pressure retaining ability will be seriously impaired.

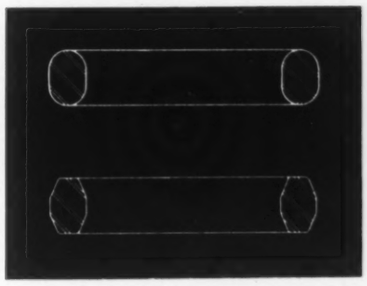


Fig. 3*a* (*above*)—Oval ring. Fig. 3*b* (*below*)—Octagonal ring

An examination of the rings after the test indicated the shape of both rings had changed from an oval section to somewhat of an octagon shape, the flattening of the oval having resulted from contact with the sides of the grooves under high bolt loads. There was no indication of bottoming of either ring and there was no other deformation of either ring than that mentioned above.

Effect of Shape of Ring on Pressure Retaining Ability of Ring Joints

In the course of conducting numerous tests on ring joints using the oval cross section rings it was found that in practically every instance the rings were deformed so as to conform to the flat surface of the flange grooves. Thus, the oval rings were virtually transformed to rings having an octagonal cross section, but this transformation was accomplished only at the expense of much time and labor spent in pulling up the bolts until such time as the ring ceased to yield and supported the desired bolt load.

With this observation in mind and with a view toward minimizing the time and labor required in the erection of large ring joints, the use of a ring of octagonal cross section which, upon installation, would conform with the flat sides of the flange grooves was suggested. Sketches of the two types of rings are shown in Figs. 3*a* and 3*b*.

Tests No. 4, 5, 6, and 8, Table 4, indicate that the pressure retaining ability of a ring joint with a ring of the octagonal shape is equal to that of an oval type ring joint. A further test was conducted using 16 in., 300 lb joints to determine whether the octagonal ring would be equal to an oval ring with respect to pressure retaining ability where the ring is large and the bolting loads are relatively small when compared with the loads which can be developed by the bolts in 1500 lb flange joints.

Three 16 in. *A.S.A.* 300 lb spools were made up and grooves for ring joints were machined into the flanges. Tests were conducted using an oval ring in one joint and an octagonal ring in the other.

Torque loads were applied to the nuts of the full threaded alloy steel bolt studs and after each increment of load the assembly was subjected to internal hydrostatic pressure until leakage at the joints occurred. It was noted in applying the bolt load that the octagonal rings made up in a much shorter length of time than the oval rings.

Table 7 shows the results obtained on hydrostatic pressure.

The results of this test indicate that joints made with octagonal rings will withstand pressures at least equivalent to those held by oval section ring joints.

Determining Width of Ring Section for Ring Joints

As has been intimated, in some cases rings are too light for the bolting load applied. Two instances of this type of failure are described in the following paragraphs and a method for determining the width of ring is explained.

Test A—20 In., 1500 Lb Ring Joint Flanged Connection

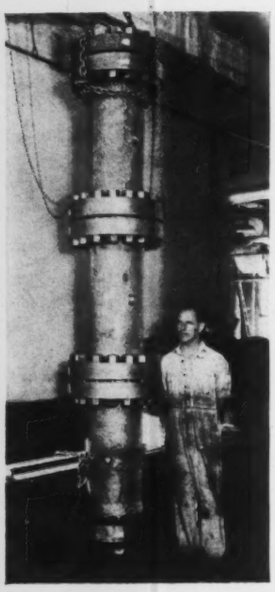
A pair of 20 in., 1500 lb lap flanges were bolted together with sixteen 3 in. alloy steel bolt studs having eight threads per inch. The flanges were made with dimensions as listed for the 20 in., 1500 lb American Standard integral flanges, *i.e.* 38¾ in. diameter x 7 in. thick. The ring used in this test was made of soft steel having a Brinell hardness of from 80 to 90 with a pitch diameter of 23 in., a width of ⅝ in., and a height of $\frac{13}{16}$ in. The joint was assembled by stressing the bolts by the thermal expansion method[1] and it was noted that after four successive increments of heating and cooling the bolts, practically no increase in bolt stress was obtained on the fourth operation over that obtained by the second. It was, therefore, obvious that the ring was distorting considerably which permitted the flanges to come together without increasing the resultant bolt load. In other words, the ring material was subjected to a compressive stress greater than its yield point.

In order to remedy this trouble, a ring was made up having a width of ⅞ in. and a pitch diameter of 23 in. The joint was reassembled and a stress of 26,000 lb per sq in. was applied to each bolt. Measurements taken of the ring before and after test indicated that only a slight change in thickness and in the height of the ring occurred due to the bolt load applied.

Test B—12 In. Drilling Through Valve—Bonnet Joint

A test was conducted on a 12 in. drilling through valve having twenty 2 in. alloy steel full threaded bonnet bolt studs. The pitch diameter and width of the ring for the bonnet joint were 21¼ and ⅝ in. respectively.

The bonnet bolts were stressed to 16,000 lb per sq in., which was the maximum obtainable due to the continuous deformation of the ring gasket.



12 in., 1350 lb *A.S.A.* flanged spool assembly—ring joint test

As it was impossible to increase the bolt load by additional turning effort on the nuts, the joint was taken apart, after the leakage pressure had been ascertained, and remachined for a ring having a pitch diameter of 21$\frac{13}{16}$ in. and a width of 1¼ in. After reassembling the joint, the bolts were pulled up to a stress of 30,000 lb per sq in. and the subsequent hydrostatic pressure test produced a result which exceeded twice the maximum pressure obtained with the thinner ring. The above tests indicate that it is imperative that rings should be so designed and proportioned to the size and area of the bolts that no collapsing or serious deformation of the ring can take place.

Table 7

Bolt Stress Applied Lb per Sq In.	Total Bolt Load Applied Lb	Oval Ring		Octagonal Ring	
		Leakage Press. Lb	Performance Factor	Leakage Press. Lb	Performance Factor
21300	395500	1500	1.00	2500	1.67
30000	557400	1950	0.92	3200	1.51
45000	836100	3200	1.01	4200	1.33

Table 8—Approximate Installation Stresses for Alloy Steel Bolt Studs Due to Manually Applied Torque Loads

Size of bolt Stud, In.	Installation Stress, Lb per Sq In.
¾	52,000
⅞	48,000
1	45,000
1⅛	42,500
1¼	40,000
1⅜	38,000
1½	36,500
1⅝	35,000
1¾	34,000
1⅞	33,000
2	32,000
2¼	30,000
2½	28,500
2¾	27,000
3	26,000

Table 9

	A	B	C	D	D÷A
	Root Area of Bolts Sq In.	Width of Ring, In.	Pitch Diam. of Ring, In.	Width×Pitch Circumference, Sq In.	Ratio
20 In., 1500 Lb. Flanges	101.18	⅝	23	45.2	0.45
	101.18	⅞	23	63.3	0.63
12 In. Drilling Through Valve	53.0	⅝	21¼	41.7	0.79
	53.0	1¼	21$\frac{13}{16}$	86.3	1.63

Table 9a

Size of Bolt, In.	Pitch Circumference × Width of Ring ÷Total Cross Sectional Area of Bolts
⅞ and smaller	2 or more
1	1.8
1⅛	1.6
1¼	1.4
1⅜	1.2
1½ to 1¾	1.0
1⅞	0.9
2	0.85
3	0.80

In determining the proper size of ring to use, the bolt load or bolt stress which presumably is applied when various sized flanged joints are assembled should be considered. For example, the installation stress applied to a 2 in. bolt will probably be much lower than that applied to a 1 in. bolt due to the fact that it is much more difficult to supply the high torque loads necessary for a 2 in. bolt to produce stresses that will be equal to that set up in a 1 in. bolt with a much lower manually applied torque. From experience it has been indicated that installation stresses due to torque loads for various sizes of alloy steel full threaded bolt studs made in accordance with the 8 pitch thread series in sizes larger than 1 in. are approximately equal to those determined by the use of the empirical formula:

$$S = \frac{45{,}000}{\sqrt{d}}$$

Table 10—Ring Dimensions for 150 Lb *A. S. A.* Flanges

Size of Flange In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Pitch Circumference × Width (1)	No. & Size of Bolts	Total Bolt Area Sq In. (2)	Ratio (1)/(2)
1	1 7/8	5/16	1.84	4— 1/2	0.504	3.65
1 1/4	2 1/4	5/16	2.21	4— 1/2	0.504	4.38
1 1/2	2 9/16	5/16	2.50	4— 1/2	0.504	4.96
2	3 1/4	5/16	3.29	4— 5/8	0.808	4.07
2 1/2	4	5/16	3.92	4— 5/8	0.808	4.85
3	4 1/2	5/16	4.42	4— 5/8	0.808	5.47
3 1/2	5 5/16	5/16	5.09	8— 5/8	1.616	3.15
4	5 7/8	5/16	5.76	8— 5/8	1.616	3.57
5	6 5/8	5/16	6.62	8— 3/4	2.416	2.74
6	7 5/8	5/16	7.48	8— 3/4	2.416	3.10
8	9 3/4	5/16	9.57	8— 3/4	2.416	3.96
10	12	5/16	11.78	12— 7/8	5.032	2.34
12	15	5/16	14.72	12— 7/8	5.032	2.93
14	15 5/8	5/16	15.35	12—1	6.612	2.32
16	17 7/8	5/16	17.57	16—1	8.816	2.00
18	20 3/8	5/16	20.00	16—1 1/8	11.648	1.72
20	22	5/16	21.60	20—1 1/8	14.56	1.49
24	26 1/2	5/16	26.05	20—1 1/4	18.58	1.40

Table 11—Ring Dimensions for 300 Lb *A. S. A.* Flanges

Size of Flange In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Pitch Circumference × Width (1)	No. & Size of Bolts	Total Bolt Area Sq In. (2)	Ratio (1)/(2)
2	3 1/4	7/16	4.47	8— 5/8	1.616	2.77
2 1/2	4	7/16	5.50	8— 3/4	2.416	2.28
3	4 7/8	7/16	6.37	8— 3/4	2.416	2.64
3 1/2	5 5/16	7/16	7.13	8— 3/4	2.416	2.96
4	5 7/8	7/16	8.08	8— 3/4	2.416	3.35
5	7 1/8	7/16	9.80	8— 3/4	2.416	4.05
6	8 5/16	7/16	11.40	12— 3/4	3.624	3.15
8	10 3/8	7/16	14.60	12— 7/8	5.032	2.90
10	12 3/4	7/16	17.52	16—1	8.816	1.99
12	15	7/16	20.60	16—1 1/8	11.648	1.77
14	16 1/2	7/16	22.65	20—1 1/8	14.56	1.56
16	18 1/2	7/16	25.40	20—1 1/4	18.58	1.36
18	21	7/16	28.85	24—1 1/4	22.296	1.29
20	23	1/2	36.10	24—1 1/4	22.296	1.57
24	27 1/4	5/8	53.50	24—1 1/2	33.72	1.58

Table 12—Ring Dimensions for 400 Lb *A. S. A.* Flanges

Size of Flange In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Pitch Circumference × Width (1)	No. & Size of Bolts	Total Bolt Area Sq In. (2)	Ratio (1)/(2)
4	5 7/8	7/16	8.08	8— 7/8	3.352	2.41
5	7 1/8	7/16	9.80	8— 7/8	3.352	2.92
6	8 5/16	7/16	11.40	12— 7/8	5.032	2.27
8	10 5/8	7/16	14.60	12—1	6.612	2.21
10	12 3/4	7/16	17.52	16—1 1/8	11.648	1.50
12	15	7/16	20.60	16—1 1/4	14.864	1.39
14	16 1/2	7/16	22.65	20—1 1/4	18.58	1.22
16	18 1/2	7/16	25.40	20—1 3/8	23.10	1.10
18	21	7/16	28.85	24—1 3/8	27.72	1.04
20	23	1/2	36.10	24—1 1/2	33.72	1.07
24	27 1/4	5/8	53.50	24—1 3/4	47.52	1.12

Table 13—Ring Dimensions for 600 Lb *A. S. A.* Flanges

Size of Flange In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Pitch Circumference × Width (1)	No. & Size of Bolts	Total Bolt Area Sq In. (2)	Ratio (1)/(2)
1/2	1 11/32	1/4	1.05	4— 1/2	0.504	2.08
3/4	1 11/16	5/16	1.66	4— 5/8	0.808	2.05
1	2	5/16	1.97	4— 5/8	0.808	2.43
1 1/4	2 3/8	5/16	2.33	4— 5/8	0.808	2.88
1 1/2	2 11/16	5/16	2.64	4— 3/4	1.208	2.19
2	3 1/4	7/16	4.47	8— 5/8	1.616	2.77
2 1/2	4	7/16	5.50	8— 3/4	2.416	2.28
3	4 7/8	7/16	6.37	8— 3/4	2.416	2.64
3 1/2	5 5/16	7/16	7.13	8— 7/8	3.352	2.13
4	5 7/8	7/16	8.08	8— 7/8	3.352	2.41
5	7 1/8	7/16	9.80	8—1	4.408	2.22
6	8 5/16	7/16	11.40	12—1	6.612	1.72
8	10 5/8	7/16	14.60	12—1 1/8	8.736	1.67
10	12 3/4	7/16	17.52	16—1 1/4	14.864	1.18
12	15	7/16	20.60	20—1 1/4	18.58	1.11
14	16 1/2	7/16	22.65	20—1 3/8	23.10	0.98
16	18 1/2	7/16	25.40	20—1 1/2	28.10	0.91
18	21	7/16	28.85	20—1 5/8	33.60	0.86
20	23	1/2	36.10	24—1 5/8	40.32	0.89
24	27 1/4	5/8	53.50	24—1 7/8	55.30	0.96

Table 14—Ring Dimensions for 900 Lb *A. S. A.* Flanges

Size of Flange In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Pitch Circumference × Width (1)	No. & Size of Bolts	Total Bolt Area Sq In. (2)	Ratio (1)/(2)
3	4 7/8	7/16	6.37	8— 7/8	3.352	1.90
3 1/2	5 5/16	7/16	7.13	8—1	4.408	1.62
4	5 7/8	7/16	8.08	8—1 1/8	5.824	1.39
5	7 1/8	7/16	9.80	8—1 1/4	7.432	1.32
6	8 5/16	7/16	11.40	12—1 1/8	8.736	1.31
8	10 5/8	7/16	14.60	12—1 3/8	13.86	1.05
10	12 3/4	7/16	17.52	16—1 3/8	18.48	0.95
12	15	7/16	20.60	20—1 3/8	23.10	0.89
14	16 1/2	5/8	32.40	20—1 1/2	28.10	1.15
16	18 1/2	5/8	36.30	20—1 5/8	33.60	1.08
18	21	3/4	49.40	20—1 7/8	46.08	1.07
20	23	3/4	54.10	20—2	53.04	1.02
24	27 1/4	1	85.61	20—2 1/2	85.84	1.00

Table 15—Ring Dimensions for 1500 Lb *A. S. A.* Flanges

Size of Flange In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Pitch Circumference × Width (1)	No. & Size of Bolts	Total Bolt Area Sq In. (2)	Ratio (1)/(2)
1	2	5/16	1.97	4— 7/8	1.676	1.18
1 1/4	2 3/8	5/16	2.33	4— 7/8	1.676	1.39
1 1/2	2 11/16	5/16	2.64	4—1	2.204	1.20
2	2 3/4	7/16	5.15	8— 7/8	3.352	1.54
2 1/2	4 1/4	7/16	5.83	8—1	4.408	1.32
3	5 3/8	7/16	7.38	8—1 1/8	5.824	1.27
3 1/2	5 7/8	7/16	8.07	8—1 1/8	5.824	1.38
4	6 3/8	7/16	8.76	8—1 1/4	7.432	1.18
5	7 5/8	7/16	10.48	8—1 1/2	11.24	0.93
6	8 5/16	1/2	13.05	12—1 3/8	13.86	0.94
8	10 5/8	5/8	20.85	12—1 5/8	20.16	1.03
10	12 3/4	5/8	25.05	12—1 7/8	27.64	0.91
12	15	7/8	41.25	16—2	42.30	0.97
14	16 1/2	1	51.84	16—2 1/4	54.70	0.95
16	18 1/2	1 1/8	65.40	16—2 1/2	68.60	0.95
18	21	1 1/8	74.10	16—2 3/4	84.00	0.88
20	23	1 1/4	90.25	16—3	101.00	0.89
24	27 1/4	1 3/8	117.50	16—3 1/2	139.20	0.85

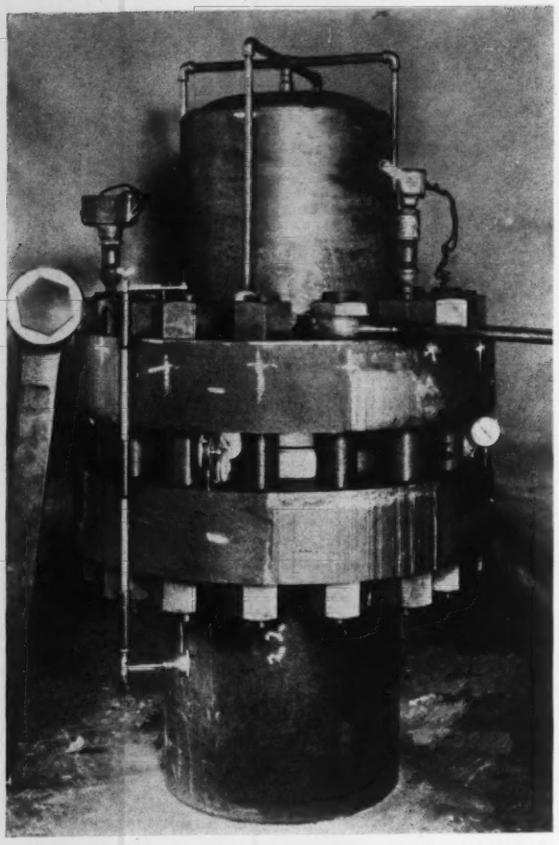
Table 16—Ring Dimensions for Proposed 2500 Lb *A.S.A.* Flanges

Size of Flange, In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Pitch Circumference × Width (1)	No. & Size of Bolts	Total Bolt Area Sq In. (2)	Ratio (1)/(2)
1	2 3/8	5/16	2.33	4— 7/8	1.676	1.39
1 1/4	2 27/32	7/16	3.90	4—1	2.204	1.77
1 1/2	3 1/4	7/16	4.47	4—1 1/8	2.912	1.54
2	4	7/16	5.50	8—1	4.408	1.25
2 1/2	4 3/8	1/2	6.87	8—1 1/8	5.824	1.18
3	5	1/2	7.85	8—1 1/4	7.432	1.06
4	6 3/16	5/8	12.13	8—1 1/2	11.24	1.08
5	7 1/2	3/4	17.65	8—1 3/4	15.84	1.11
6	9	3/4	21.20	8—2	21.15	1.00
8	11	7/8	30.24	12—2	31.73	0.95
10	13 1/2	1 1/8	47.70	12—2 1/2	51.45	0.93
12	16	1 1/4	62.80	12—2 3/4	63.00	0.99

Table 17—Ring Dimensions for Oil Field (Production) Flanges

Size of Flange, In.	Pitch Diameter of Ring, In.	Width of Ring, In.	Ratios[a]				
			500 Lb W. P. (300 Lb A. S. A.)	1000 Lb W. P. (400 Lb A. S. A.)	2000 Lb W. P. (600 Lb A. S. A.)	3000 Lb W. P. (900 Lb A. S. A.)	5000 Lb W. P. (1500 Lb A. S. A.)
1	2	5/16	....	....	2.43	....	1.18
1 1/4	2 3/8	5/16	....	....	2.88	....	1.39
1 1/2	2 11/16	5/16	....	....	2.19	....	1.20
2	3 1/4	7/16	2.77	....	2.77	....	1.33
2 1/2	4	7/16	2.28	....	2.28	....	1.25
3	4 7/8	7/16	2.64	....	2.64	1.90	1.09
3 1/2	5 5/16	7/16	2.96	....	2.13	1.62	1.2
4	5 7/8	7/16	3.35	2.41	2.41	1.39	1.09
5	7 1/8	7/16	4.05	2.92	2.22	1.32	0.87
6	8 5/16	7/16	3.15	2.27	1.72	1.31	0.82
8	10 5/8	7/16	2.90	2.21	1.67	1.05	0.72
10	12 3/4	7/16	1.99	1.50	1.18	0.95	0.63
12	15	7/16	1.77	1.39	1.11	0.89	....
14	16 1/2	7/16	1.56	1.22	0.98	....	....
16	18 1/2	7/16	1.36	1.10	0.91	....	....
18	21	7/16	1.29	1.04	0.86	....	....
20	23	1/2	1.57	1.07	0.89	....	....

[a]Ring cross sectional area ÷ total cross sectional area of bolts.



Test set-up for 20 in., 1500 lb cast steel flanged ring joint showing heaters and instruments in place

where S = Installation stress, lb per sq in.
d = Nominal diameter of bolt stud.

Table 8 gives installation stress values for various sizes of bolt studs.

As the ring must withstand the bolt stresses applied to it in service, it is apparent from Table 8 that the proportion of the size of ring to the bolt area can be lower for the larger sizes of bolts than for the smaller sizes. Therefore, in order to determine the size of ring for a definite size of flange, the following empirical method is proposed:

The pitch circumference of the ring should be multiplied by the width of the ring cross section. This will give the horizontal cross sectional area of the ring. The ratio between this figure and the total cross sectional area of the bolts at the root of thread should then be obtained and compared with the recommended values given in Table 9a.

Table 9 shows the relationship between the rings and bolts used in Tests *A* and *B* just described.

It is noted from Table 9 that rings yielding ratios of less than approximately 0.65 on the 20 in. and 0.85 on the 12 in. flanges would be subjected to stresses beyond the yield point of the material from which they are made when the joints are assembled with normal bolt stresses. On the basis of these tests and also an analysis of all rings heretofore furnished which have proved to be satisfactory, Table 9a is proposed to indicate the approximate ratios which should be used in determining dimensions of rings for ring joints.

Application of Ring Joints

The dimensions of rings for ring joints, as given in Tables 10 to 17 inclusive have been proposed by an *A.P.I.* committee for a ring joint flange standard. The ratio of ring cross sectional area to bolt area is also tabulated. Tables 10 to 16 give dimensions of rings for all services except oil field production, and Table 17 gives ring dimensions for production flanges.

The author gratefully acknowledges the valuable assistance of the personnel of the Industrial Laboratory[2] and also the co-operation of A. M. Houser, Engineer of Standardization,[2] in the preparation of this article.

[2]Of the Division of Research and Development, Crane Co.

Air Conditioning Load a Reason for New Gas Rate

OF interest as it concerns motive power for air conditioning is the new "gas motor service rate" announced last month by The Peoples Gas, Light and Coke Co., Chicago, which marks the entry of the gas company into the competitive field of industrial power supply. A primary consideration in establishing the rate was the gas company's belief that it would lead to the wide application of internal combustion gas engines for driving compressors for air conditioning and ice-making plants, as the terms are particularly attractive during the summer months.

The company's natural gas contract provides for payment of a capacity charge established by peak demands, and a commodity charge which varies with the amount of natural gas actually taken from the pipeline from day to day. Development of space heating gas sales has left the company a substantial amount of gas during the non-heating season which it can afford to sell at low rates to balance summer and winter load. The rate is therefore designed to encourage the maximum use of gas for industrial power in the April to November period.

Terms are 3c per therm (a therm is 100,000 Btu) for the first 1500 therms of gas used in any month from May 1 to September 30, dropping to 2.25c per therm for all gas in excess of that amount used in any month during this period; 4.25c per therm for gas used in April and October and 12c per therm for gas used in gas motors between November 1 and March 31. Two 400 hp gas motors now being installed to generate electricity for industrial use are designed to burn fuel oil in the winter months.

How to Avoid Trouble on Cooling Jobs With Evaporators Above Condensers

[Part 1]

By William Goodman*

Where it is necessary to place the evaporating coils some distance above the condenser or receiver—as it is in many air conditioning installations—problems are created which must be recognized and properly dealt with if trouble is to be avoided. Among the difficulties are those due to incorrect selection of the expansion valve. . . . Here is a thorough discussion of the problem involved and an explanation of what to do about it

WHERE it is necessary in an air conditioning or refrigerating installation to place the evaporating coils some distance above the condenser or receiver, problems are created which must be recognized and properly dealt with if subsequent difficulties are to be avoided.

Among the principal difficulties are those due to the incorrect selection of expansion valves. Valves must be chosen not only for the smaller pressure difference, due to the lower pressure existing at the top of vertical pipe lines, but their selection must also allow for the small amount of vapor formed in the vertical line. Although the actual weight of vapor is small, the volume occupied by it is relatively large—so much so that it has a substantial effect upon the capacity of an expansion valve.

Pressure in a Vertical Pipe Line

As in all vertical liquid columns, the pressure at any point depends upon the height of the column above it. Thus, if the pressure at the base of a vertical pipe line of water is 100 lb per sq in, the pressure in this line will fall off 1 lb per sq in. for every 2.3 ft of vertical distance above the base. At a distance of 23 ft above the base, the pressure will be 10 lb less, or 90 lb per sq in. In the same way, the pressure in a vertical pipe line full of liquid "Freon" varies, being highest at the base and lowest at the top.

In a vertical pipe line full of water, the density of the water is substantially the same from top to bottom. This is not true in a pipe line of "Freon" because a portion of the "Freon" vaporizes as the pressure falls in the vertical line. To understand this, it is necessary to refer to Table 1, which is a reproduction of a portion of the *A. S. R. E.* "Tables of the Properties of 'Freon.'" The second and fourth columns show the *lowest* pressure at which 1 lb of "Freon" can remain entirely liquid at any given temperature. If the pressure is lowered below the tabulated value, the liquid will immediately cool to the temperature corresponding to the new low pressure.

For example, if the pressure of liquid "Freon" at a temperature of 90 F is lowered from, say, an initial pressure of 110 lb ga to a final pressure of 102 lb ga, there will be no change in the temperature of the liquid, for (referring to Table 1) at a temperature of 90 deg the corresponding pressure is 99.6 lb ga. However, if the pressure is lowered still further to, say, 81.2 lb ga, the liquid will immediately cool to 78 F, the corresponding temperature given in Table 1. In cooling from 90 to 78 F, the liquid "Freon" surrenders heat, which heat vaporizes a small part of the liquid itself. In the preceding illustration, if the pressure of the liquid remains above 99.6 lb ga, there will be no vaporization. However, if the pipe line is of sufficient height so that the pressure falls below 99.6 lb ga, a portion of the liquid "Freon" will vaporize inside the liquid line. The amount that will vaporize depends entirely upon the final pressure—the lower the final pressure, the greater the weight that will vaporize. The action of the "Freon" in the vertical pipe line is the same as its action when it flows through an expansion valve; a portion of it must flash into vapor due to the reduction in pressure.

As long as a portion of the liquid line is insulated (to

Table 1—Saturation Pressures of "Freon"
(From A.S.R.E. Data)

Temperature Deg Fahr	Gage Pressure Lb per Sq In.	Temperature Deg Fahr	Gage Pressure Lb per Sq In.
30	28.5	80	84.1
32	30.1	82	87.0
34	31.7	84	90.1
36	33.4	86	93.2
38	35.2	88	96.4
40	37.0	90	99.6
42	38.8	92	103.0
44	40.7	94	106.3
46	42.7	96	109.8
48	44.7	98	113.3
50	46.7	100	116.9
52	48.8	102	120.6
54	50.9	104	124.3
56	53.1	106	128.1
58	55.4	108	132.1
60	57.7	110	136.0
62	60.1	112	140.1
64	62.5	114	144.2
66	65.0	116	148.4
68	67.5	118	152.7
70	70.1	120	157.1
72	72.8	122	161.5
74	75.5	124	166.1
76	78.3	126	170.7
78	81.2	128	175.4

*The Trane Co., LaCrosse, Wis., Member of Board of Consulting and Contributing Editors.
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be discussed later) there is no loss in refrigerating capacity due to the flashing of the vapor in the liquid line. As the liquid flows through the expansion valve, a fixed portion of it must vaporize in any event. This being the case, it makes no difference—insofar as the refrigerating capacity of the system is concerned—whether all of this vaporization takes place in the expansion valve, or part in the liquid line and the balance at the expansion valve. However, the capacity of the expansion valve itself is affected, as will be discussed later.

In analyzing the action in any vertical pipe line, it is necessary to divide the line into three zones, Fig. 1. In the first zone there is no vaporization of the liquid. Vaporization of the liquid cannot take place until the pressure in the liquid line has fallen to a point equal to the pressure corresponding to the temperature of the liquid as given in Table 1. When liquid "Freon" leaves a condenser, it is at a temperature equal to or slightly lower than the temperature at which it was condensed. For instance, if "Freon" vapor is condensing at a pressure of 116.9 lb ga, its condensing temperature will be, from Table 1, 100 F. If the liquid is then cooled to 90 F after leaving the condenser, no vaporization can take place in the vertical line until the pressure falls below 99.6 lb ga. The point on the vertical pipe line where the pressure is equal to the pressure corresponding to the temperature of the liquid (as given in Table 1) will be called, for convenience, the *lower boundary*. It represents the boundary line between the solid liquid in the pipe line and the point at which bubbles of vapor begin to form in the liquid, and in Fig. 1 is represented by the line *C-D*. A liquid zone can exist only if the liquid "Freon" is subcooled after condensation takes place; that is, the refrigerant is cooled to a temperature below that at which it was condensed. If liquid enters the liquid line at the same temperature at which it was condensed, there will be no liquid zone and the line *C-D* will move down until it coincides with the line *A-B* representing the liquid level in the receiver.

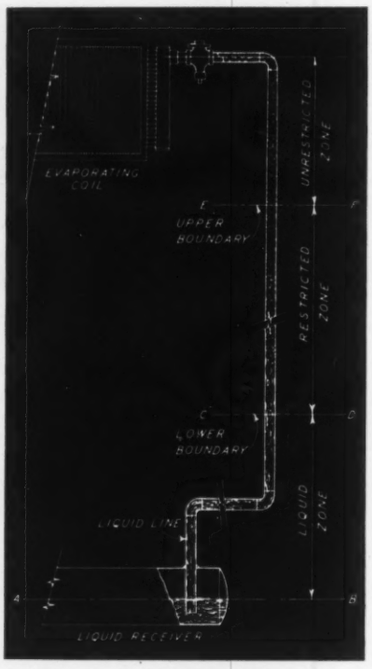


Fig. 1—Where the evaporator is some distance above the condenser or receiver, as illustrated here, problems are created which must be recognized if difficulties are to be avoided, as is explained

The next zone in Fig. 1 is called the *restricted zone* because it represents the one in which vaporization of the liquid is restricted; why this is so will be apparent from the following discussion.

The amount of liquid vaporizing depends entirely upon the lowest pressure to which it is subjected—the lower the pressure, the greater the quantity of liquid that will vaporize. If the pressure falls off in the liquid line and the liquid vaporizes, the remaining liquid "Freon" cools; not only the pressure but the temperature of the "Freon" liquid falls off progressively from the lower boundary to the highest point on the pipe line. As long as the temperature of the liquid "Freon" is above the temperature of the surrounding air, there is no heat flow from the surrounding air into the liquid pipe line. Any vaporization which takes place is due entirely to the falling off of pressure, and the latent heat for this vaporization is obtained entirely from the cooling of the liquid "Freon."

However, should the pressure in the liquid line fall to such a point that the temperature of the liquid "Freon" is below the temperature of the surrounding air, then there will be a flow of heat from the surrounding air through the walls of the pipe to the liquid "Freon," which will result in further vaporization of the "Freon." It is conceivable that if there were enough bare pipe surface, all of the liquid "Freon" might vaporize before it reached the expansion valve.

The point on the pipe line at which the pressure is such that the temperature of the liquid "Freon" is equal to the temperature of the surrounding air will be called the *upper boundary*. The temperature of the liquid "Freon" at all points between the lower and upper boundaries is higher than the temperature of the surrounding air. In this restricted zone the vaporization of the liquid "Freon" can only take place because of a drop in pressure, and all of the latent heat is obtained by the cooling of the liquid.

How to Insulate the Piping

The portions of the liquid line corresponding to the liquid and restricted zones should not be insulated, but the portion of the pipe line in the unrestricted zone should be insulated if a loss in refrigerating capacity is to be avoided. It may be trite to say that refrigeration can only be obtained by vaporization of the liquid in the evaporator coils, but it seems advisable to stress that, aside from the inevitable vaporization due to the flashing of the liquid because of the pressure drop, any further vaporization that takes place is a definite and avoidable loss. The vaporization that occurs in the restricted zone cannot be considered as an unavoidable loss. It is simply a portion of the loss that would take place in the expansion valve. However, any vaporization of the liquid "Freon" that takes place in the unrestricted zone —due to heat flowing from the surrounding air—represents a definite loss in refrigerating capacity. The cooling that can take place in the evaporator will be lower by exactly the amount of heat gained by the liquid "Freon" from the air surrounding the liquid line. When all things are considered, it is best not to place evaporators or cooling coils above the upper boundary. Where this cannot be avoided, the portion of the liquid line above the upper boundary should be carefully insulated.

Computing Pressure at Any Point in Line

It is a simple matter to compute the decrease in pressure in the *liquid* zone due to the vertical rise of the pipe

line. The density of liquid "Freon" varies from 82.6 lb per cu ft at 70 F to 76.0 lb at 120 F. Considering other sources of error, an average value of 80 lb per cu ft may be used. This means that the pressure in the liquid zone will decrease 1 lb per sq in for every 1.8 ft of vertical rise (144 sq in. per sq ft divided by 80 lb per cu ft equals 1.8).

On the other hand, above the lower boundary line, due to the fact that the liquid is continuously vaporizing as it rises higher in the pipe line, the density of the mixture of liquid "Freon" and vapor is constantly decreasing. This makes it difficult to compute the pressure at any point above the lower boundary. A 1 ft column of liquid "Freon" at the lower end of a vertical pipe will cause considerably more change in pressure than will a 1 ft column of a mixture of liquid and vapor at the upper end.

For this reason, it was necessary to prepare Fig. 2, which can be used to find the final gage pressure at any vertical distance above the lower boundary line, if the pressure at the lower boundary line is known. The method used in preparing Fig. 2 will be given in the appendix.

Example 1 "Freon" vapor is condensing under a pressure of 124.3 lb ga. The liquid is subcooled 4 deg before leaving the condenser. Find the location of the lower boundary.

Solution: Referring to Table 1,

Condensing temperature of "Freon" at a pressure of 124.3 lb ga	104 deg
Subcooling	4 deg
Temperature of liquid "Freon"	100 deg

Referring to Table 1, saturation pressure of "Freon" corresponding to a temperature of 100 F is 116.9 lb ga.

Condensing pressure	124.3
Lower boundary pressure	116.9
Pressure difference	7.4 lb ga

Vertical distance between liquid receiver and lower boundary line = 7.4 × 1.8 = 13.3 ft

The location of the upper boundary line depends, of course, upon the temperature of the air surrounding the liquid pipe line. The higher the temperature of the surrounding air, the lower will be the location of the upper boundary line. Therefore, in deciding how much of the liquid line should be insulated, it is necessary to use the highest temperature to which the surrounding air is likely to rise.

Example 2: Take the conditions of Example 1 and find the location of the upper boundary line if the highest surrounding air temperature is 90 F.

Solution: Referring to Table 1, when the temperature of the liquid "Freon" has fallen to 90 F the pressure at the upper boundary will be 99.6 lb ga.

From Example 1, pressure at lower boundary = 116.9 lb ga.

Referring to Fig. 2, for a lower boundary pressure of 116.9 lb ga and a final gage pressure of 99.6 lb, the vertical distance between the lower and upper boundary lines will be 50 ft.

Distance from liquid receiver to lower boundary	13.3 ft
Distance from lower boundary to upper boundary	50.0 ft
Vertical distance from liquid receiver to upper boundary	63.3 ft

All of the liquid line between the upper boundary and the expansion valve should be insulated. Thus, in the preceding example, if the evaporator were 80 ft above the liquid receiver, it would be necessary to insulate the line for a distance of 80 minus 63.3 or 16.7 ft. If there are horizontal runs of liquid line above the upper boundary, they too must be insulated. Thus if there were 15 ft of horizontal run above the upper boundary, it would be necessary to insulate this 15 ft of horizontal run as well as the 16.7 ft of vertical line, making a total of 31.7 ft to be insulated.

Even though the portion of the liquid pipe line passing through the unrestricted zone should be insulated, the portion in the liquid and restricted zones should *not* be insulated. Any heat losses from these two lower zones will result in an increase in the capacity of the refrigerating system. In the liquid zone, heat losses will lower the liquid temperature, thus lowering the amount of liquid which will flash to vapor. In the restricted zone, heat losses will cause a portion of the flash vapor to condense, also resulting in an increase in refrigerating capacity. However, these heat losses will probably not be large enough to affect the computation methods presented here by more than a negligible percentage.

[*To be continued. The author will discuss the selection of expansion valves, with particular attention to installations where the evaporator coils are above the condenser.*]

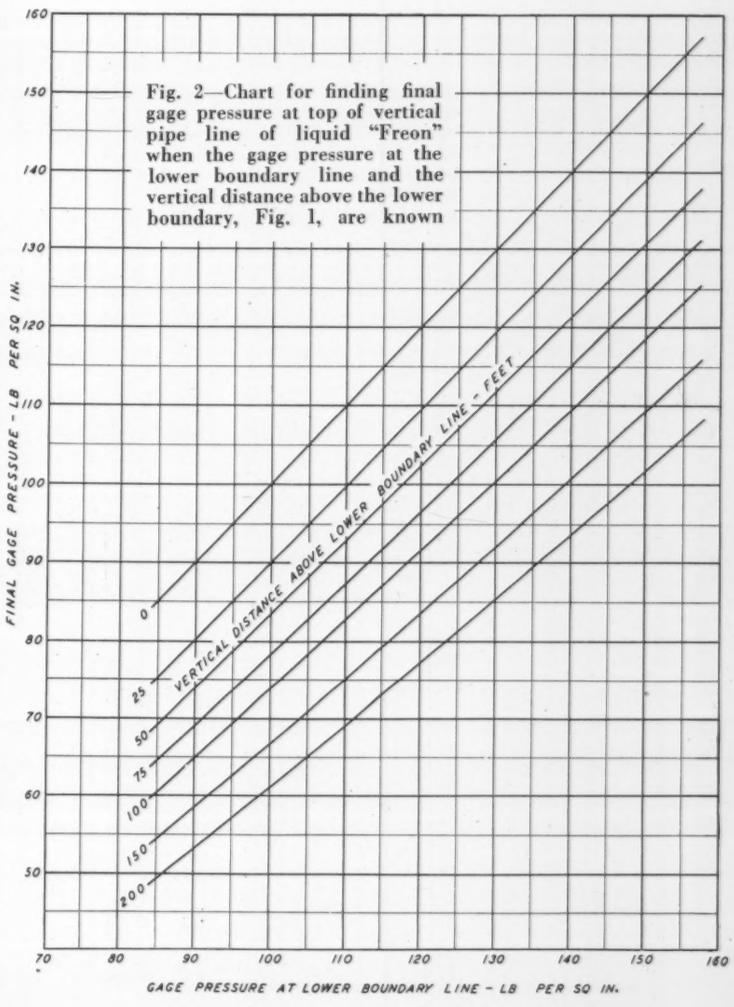


Fig. 2—Chart for finding final gage pressure at top of vertical pipe line of liquid "Freon" when the gage pressure at the lower boundary line and the vertical distance above the lower boundary, Fig. 1, are known

Superheated Water System Heats New Swiss Buildings

THE two newly constructed cantonal administrative buildings in Zurich, known as the Walche, Fig. 1, satisfy a need which has long been felt, by bringing together the various branches of the cantonal administration. In the immediate neighborhood of the two buildings is the Kaspar-Escher house, which was bought by the canton in 1920, since even at that date it was seen necessary to centralize the administration of various departments. In spite of that, many departments remained isolated, and this had not only an unfavorable influence on internal service, but also proved inconvenient to the public.

The question of heating the Kaspar-Escher house and also the two buildings was the object of special investigation. As the new buildings are only about 300 yards away from the power station of the largest district heating system hitherto constructed in Zurich, the possibility of making use of district heating instead of installing separate heating plants was taken into consideration.

Since the boiler plant in the old Kaspar-Escher house was also in a state requiring renewal, a more favorable utilization of the heat transmitting plant could be reckoned with, if also this building was changed over to district heating.

A further increase in the maximum quantity of heat to be transmitted could be taken into consideration if the premises of the insurance company "La Suisse", which is near the Walche building, were also connected. The result of the preliminary investigations showed that the cost of heating would be about the same whether the heat was supplied by the district heating or generated in a separate plant. Other considerations with regard to simpler service, etc., led to preference being given to district heating.

The considerable difference in level between the power station and the substations excluded a low pressure hot water system. Also since the existing plant in the Kaspar-Escher house was of the low pressure steam heating type, it would have been a costly matter to convert it to hot water heating. Consequently the only heat transmitting media considered were superheated water or steam. Comparative calculation showed that the transmission of heat by means of superheated water gave the most favorable solution both with regard to service and to economy.

As can be seen from Fig. 2, the district heating piping is a two pipe system. Since apparatus requiring heat at a high temperature (steam generators), and also apparatus requiring heat at a lower temperature (central heating water and service water generators), is connected to the district heating main, transformers could be coupled in series for the purpose of making the temperature of the return flow of the superheated water as low as possible. When a certain amount of heat is required from the district heating main, the excellent utilization of the superheated water allows the main to be of smaller dimensions, with a consequent reduction in the cost of installing it. Nevertheless, only the apparatus serving for room heating was coupled in series, namely the superheated water/steam transformers and the superheated water/hot water transformers. The heat requirements of these two groups of transformers are always in a certain fixed relation to each other, since they depend on the outside temperature at the moment; coupling them in series, as shown in Fig. 2, made a simpler arrangement possible. According to this arrangement, the superheated water/steam transformers are regulated by means of three-way valves, while a quantity of water, which is independent of the momentary setting of the regulating valves, flows constantly through the compensating pipe between the two groups of transformers. In this way a supply of superheated water to the second stage is insured at all times. The second stage regulation is effected in a similar manner by means of three-way valves, so that the flow of superheated water through the first stage is uninfluenced by the position of the regulating members of the second stage.

Method of Heating the Water

The superheated water is produced in a vertical spirally welded tank by the condensation of steam at an initial pressure of 140 lb per sq in.; in winter this steam is ex-

Fig. 1—View of the new cantonal administrative buildings, Zurich

By R. Ruegg*

tracted from a turbine, and in summer it is raised in electrode boilers. The tank can also be operated as a superheated water accumulator with constant flow temperature or as an accumulator with fall of pressure.

The steam is led at a pressure of 140 lb into the upper part of the accumulator and condenses on the water which is delivered by the accumulator circulating pumps into the steam space of the accumulator and flows over baffle plates. In this way the accumulator is gradually heated up to 340 F.

The superheated water is taken from the upper part of the accumulator, while the return water is led into the lower part. Consequently there is an upper layer of hot water and a lower layer of cold water, the dividing plane between the two layers tending to move upward or downward according as the accumulator is being charged or discharged. From the lower part of the accumulator, cold water is continuously being taken by the circulating pumps and delivered into the steam space. When the quantity of heat introduced is greater than the quantity withdrawn for use, the accumulator is being charged; when it is less, the accumulator is being discharged.

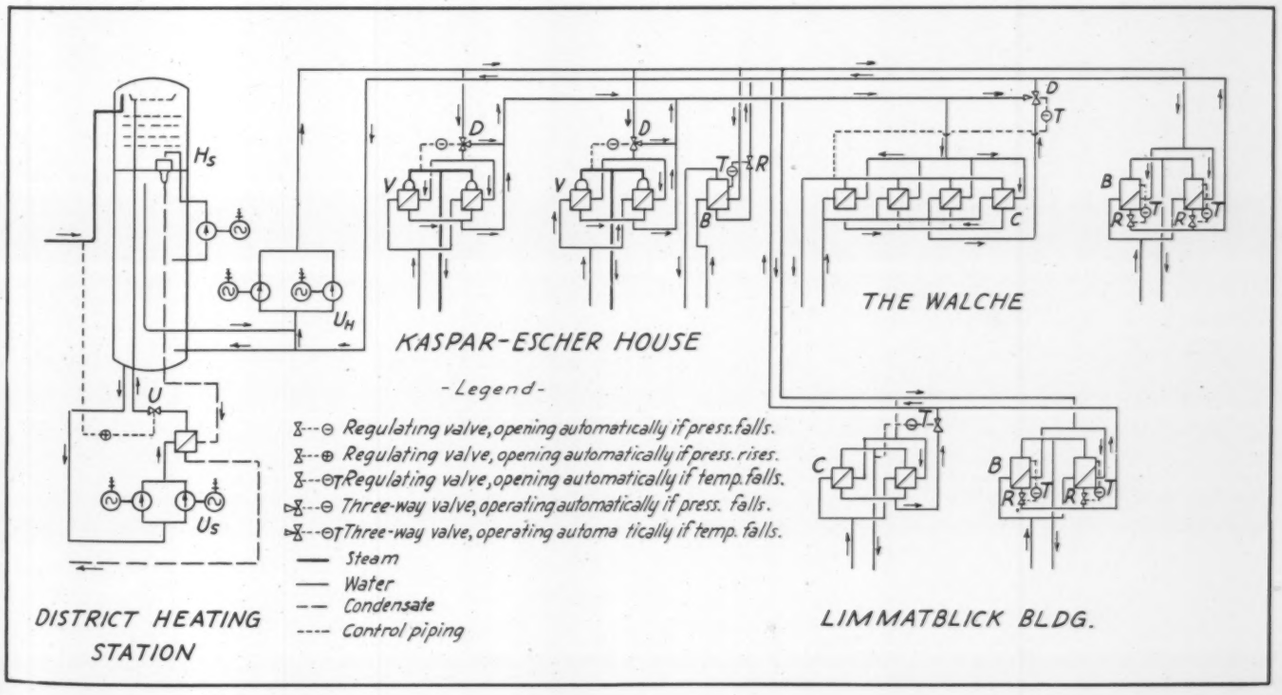
Since the quantity of steam available at 140 lb is limited at certain times, and fluctuations caused by the demand of steam by a large consumer of heat in the form of steam may consequently become unpleasantly felt, automatic regulation is provided for the steam flowing to the accumulator; this acts in such a way that the pressure in the steam main controls a valve for regulating the quantity of water circulated. When the steam pressure is falling, the regulating valve is partly or entirely closed by oil pressure, and the quantity of water cir-

Fig. 3—View of the sub-station in the Walche building, Zurich. To the right, four superheated water/hot water heat exchangers for central heating; in the background, the boilers for hot water supply, heated by superheated water; to the left in the foreground, pumps for circulating hot water

*Sulzer Brothers, Ltd., Winterthur, Switzerland.

Fig. 2—Diagrammatic arrangement of connections for supplying superheated water from district heating power station, and transforming for heating cantonal administrative buildings, Zurich

- Hs Superheated water accumulator.
- U_H External circulating pumps for superheated water.
- Us Internal circulating pumps for superheated water.
- U Regulating valve, controlled by oil under pressure.
- R Regulating valves.
- D Three-way regulating valves.
- T Control piping for regulating the temperature.
- V Heat exchangers, superheated water/steam.
- B Heat exchangers, superheated water/hot water supply.
- C Heat exchangers, for central heating.



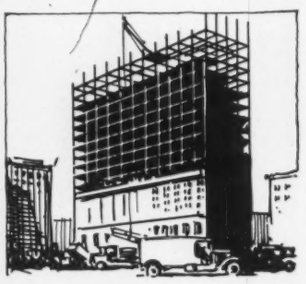
culated, and thereby also the quantity of the condensed 140 lb steam, is reduced until the pressure in the steam system has again reached the proper figure. When the pressure in the 140 lb steam main is rising, regulation is effected in the contrary sense.

In consequence of a special device, the accumulator can work with drop of pressure, the upper part of the volume of water being constantly heated to the temperature of saturated steam. The device consists of a small circulating pump, built into the upper part of the accumulator and circulating a constant quantity of water. If the pressure in the 140 lb main suddenly falls, the valve regulating the circulation is first of all closed. If the pressure falls still further, a certain quantity of steam, depending on the extent to which the pressure falls, can be given up by the accumulator to the steam main.

The Distribution System

In order to insure a minimum loss of heat in the district heating main, the flow temperature of the superheated water is adjusted to suit the quantity of heat required at the moment. The maximum flow temperature is 340 F, while the minimum permissible temperature is 265 F, because of the necessity of generating steam in the Kaspar-Escher house. The desired flow temperature is adjusted as required by an oil operated mixing valve, which allows cold water to pass from the return main to the flow main.

The energy absorbed by the external superheated water pump is regulated according to requirements; for this purpose the pumps are equipped with three-phase motors, the poles of which can be varied to give speeds between 2800 and 1430 rpm with an output of 35 or 7 hp respectively. This arrangement allows the pumps to be adapted to requirements to a large extent, particularly for summer service, when they are used only to deliver superheated water for the service water supply.



The maximum pressure occurring in the district heating main, owing to the difference in levels and the pressure of the pump, is about 350 lb per sq in. Owing to the use of expansion joints designed for this high pressure, and also by supporting the pipe on rollers which eliminate sliding friction, and owing also to the sections of the pipe being connected by a patented method of welding, it has been possible to place the pipes in inaccessible underground ducts and thus reduce the cost of installing them.

In order still further to reduce the cost of constructing the district mains, the pipes have been laid as far as possible in basements. The manner in which the whole of the main is laid is approximately as follows:

42 per cent in the basements of buildings;
7 per cent in ducts with accessible passageway
15 per cent in ducts through which a man can creep;
36 per cent in inaccessible ducts.

The district heating main is designed for transmitting 24,000,000 Btu per hr. This is considerably more than is required by the buildings at present connected; it will consequently be possible at a later date to connect several other buildings which are situated in the immediate neighborhood.

The Building Heating Systems

As can be seen from Fig. 2, in accordance with the design of the existing heating installation in the Kaspar-Escher house, two sub-stations have been installed, each with two superheated water/steam transformers, taking the place of the former coke fired steam boilers. By fitting a heating coil, the existing boiler plant has been converted to be heated by superheated water.

In the new administrative buildings, hot water was chosen for the central heating medium. In the central substation, which has been installed in the larger of the two buildings, four superheated water/hot water transformers were therefore installed. The hot water is circulated in the buildings by means of low lift centrifugal pumps. In addition, two boilers, heated by superheated water, were provided for the service water supply. Fig. 3 shows the interior of this substation.

The substation of the administrative building "Limmatblick" is arranged in a similar manner; owing to the space available, horizontal boilers could be installed there.

Charging for Heat

The heat is charged for by measuring, at the entrance to the Kaspar-Escher house, the total quantity of heat supplied to the main from the district heating power station of the Federal Technical University. Further, the heat required by the administrative buildings "Limmatblick" is measured by a heat meter. The difference between the quantities of heat thus determined is the amount required by the cantonal administrative buildings. The quantity of heat to be paid for by the canton to the suppliers is charged according to a double tariff, consisting of a fixed base charge and a charge depending on the quantity of heat consumed. The base charge is independent of the heat used, since it corresponds to the fixed charges incurred by the suppliers of heat through having to keep the necessary quantity of heat always available. The price for the quantity of heat consumed is calculated from time to time in accordance with the price of coke in Zurich. For every 1000 Btu supplied, the sum to be paid is the price of 0.31 lb of coke.

This method of charging for the heat insures that the price is always reasonable and in accordance with prevailing conditions. It takes account not only of the usual fluctuations in the price of fuel, but also of the different conditions caused by the quantity of heat required varying according to climatic conditions.

Theater air conditioning systems will pay for themselves in five years with only a four per cent patronage increase, L. Logan Lewis* told the *Motion Picture Theater Owners Association* meeting at Miami last month. He emphasized that it always costs more to condition an existing house than it does a building under construction.

*Chief Engineer, Carrier Corp., Newark, N. J.

University of Illinois Installs New Experimental Conditioning Equipment

By William H. Severns* and Maurice K. Fahnestock**

THE installation of experimental air conditioning equipment has been recently completed in the mechanical engineering laboratory at the University of Illinois and is now in service for instructional purposes. This modern all-year plant is available not only for the use of students in undergraduate and graduate courses dealing with air conditioning problems, but also may be used for air conditioning research projects.

An attempt has been made to incorporate more features in a single assemblage of apparatus than are found in the usual commercial installation. Space has been provided between all of the units for the insertion of instruments for demonstrating and studying the performance of any one part of the plant. These provisions, and the incorporation of special features, such as three separate methods of cooling and dehumidifying air, make the overall length of the plant several times greater than that of the usual installation of the same capacity.

The new experimental air conditioning installation recently completed at the University of Illinois is available not only for the use of undergraduate and graduate students, but for research work as well. So that all manner of tests and studies may be made, the design incorporates a great many features not found in the ordinary commercial installation. . . . This thorough description of the layout will interest all concerned with training and research in air conditioning, and institutions and industrial organizations contemplating similar installations.

At present no attempt is made to condition the air of a given space; the equipment is designed to handle either outdoor air, recirculated air from the main laboratory, or both, with its discharge into the laboratory. This laboratory is quite large (135 by 120 ft) and the operation of the equipment has but little effect upon the air conditions in the space. Future plans include a system of supply and return ducts between the plant and a 100 seat lecture room and several adjoining rooms. Thus, a more suitable and satisfactory means of placing a load on the plant will be afforded. Air conditioning of the additional smaller enclosed and occupied spaces will materially increase the usefulness of the equipment and broaden the scope of possible work to include studies involving the effect of air temperature, velocity, and register or diffuser locations on air distribution.

Normal cooling capacity of the equipment is equivalent to 12 tons of refrigeration when handling 3000 cfm of air, which is more than adequate to cool and dehumidify, in hot weather, the air for the 100 seat lecture room when it is fully occupied. For heating, the extended surface preheater coil has a capacity of 162,000 Btu per hr when the temperature of the entering air is 0 F and the reheater is capable of adding 140,000 Btu per hr to the air when it is received at 50 F. All capacities are based on the circulation of 13,500 lb of dry air per hr and 5 lb steam pressure in the coils.

The side elevation of the air handling parts shows the outdoor air duct; the recirculating air duct; the air filter section; the air preheater coil and bypass dampers; the atomizing mist nozzles for regulating the humidity of air to be conditioned at any season of the year; the air cooling and dehumidifying coil using chilled water as a cooling medium (this coil may be used with heated water under forced circulation for warming air); the air cooling and dehumidifying coil employing direct expansion of "Freon"; the air washer which may be used with either recirculated, heated or refrigerated spray water depending upon whether humidification or dehumidification is desired; the control dampers at the washer outlet; the bypass connection between the recirculating duct and the main duct near the air washer outlet; the air reheater coil; the backward curved blade fan; and the air discharge duct. Beneath the cooling coils are situated the water heater and the water pump; the latter may handle either recirculated, heated, or chilled water. One side of the main air duct has a number of removable access doors fitted with glass windows. Within the duct sections, where access doors having windows are placed, are electric lights. The temperature and humidity controls located in the main air duct are enumerated later. The plan view shows the arrangement of compressor, condenser, water cooler and the main assembly of the air handling units.

Restricted Duct Sections Aid Determining Velocities

Attention is called to the restricted sections of the recirculating, the outside air and the discharge air ducts, the purpose of which is to permit determination of air velocities with smaller percentages of error than is possible where the cross-sectional areas are greater. The duct sections with reduced cross-sectional areas simulate venturi tubes and do not add greatly to the total resistance against which the fan must work. The larger cross-

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sectional areas of the outside air, the recirculating air and the discharge air ducts are designed for a maximum air velocity of 1000 fpm based on air at 70 F and a barometric pressure of 29.92 in. of mercury.

Damper and Temperature Controls

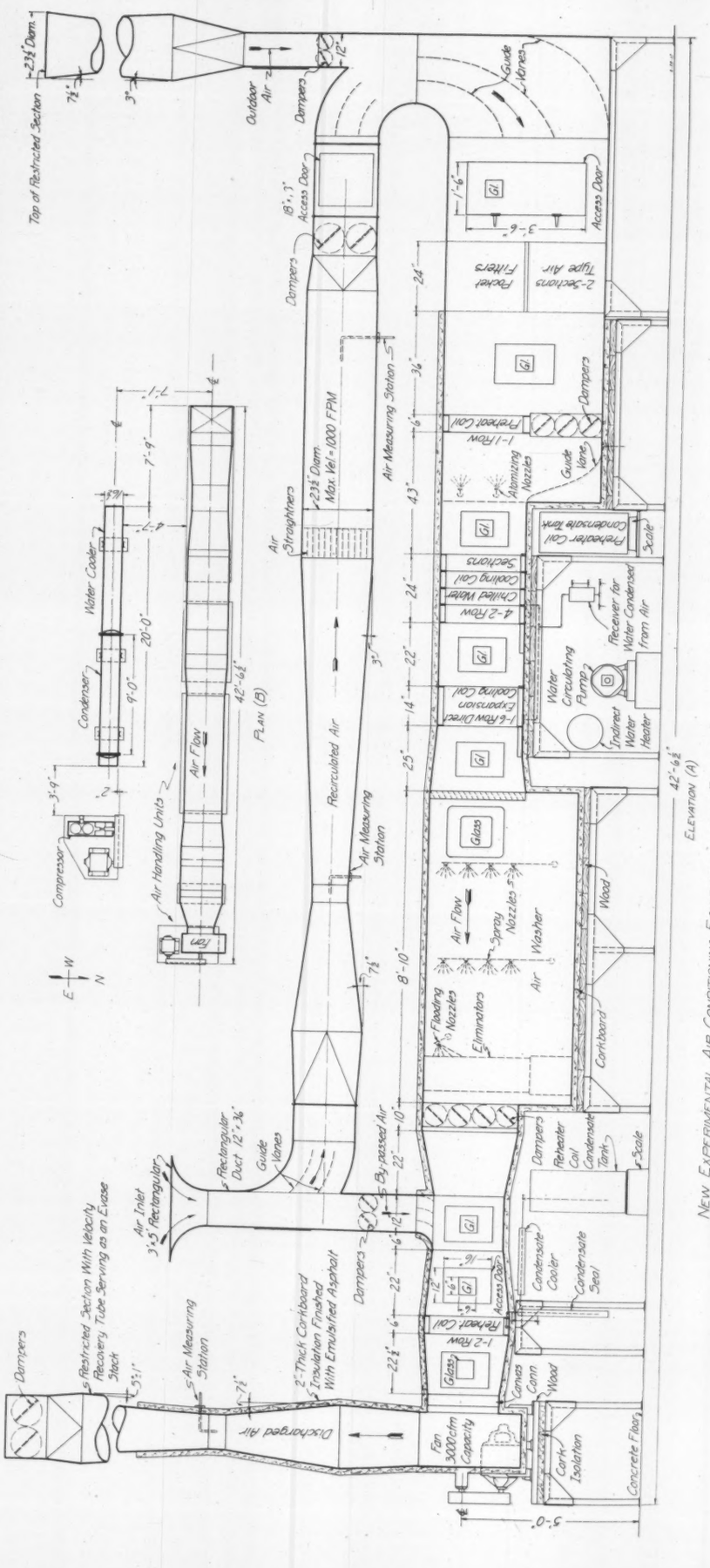
All the air dampers are manually operated at present. Air temperature and humidity control are effected by means of duct thermostats and hygrostats which operate pneumatic steam and water metal diaphragm valves using 15 lb compressed air.

The air preheater and reheater and the water heater steam supply lines are fitted with direct acting modulating valves. A gradual acting, extended disc thermostat ahead of the preheater coil functions to supply steam to the preheater when the entering air temperature drops to 35 F or less. A gradual acting duct thermostat between the atomizing mist nozzles and the air cooling coils also functions to operate the modulating valve controlling the steam supply to the preheater coil. Independent control of the preheater modulating valve is secured by having the two duct thermostats work through a pilot valve.

The temperature of the water leaving the water heater is dependent upon the adjustment of a gradual acting, immersion type thermostat in the line leading away from the heater. This thermostat or regulator operates the modulating valve in the steam supply line to the water heater.

A gradual acting, remote bulb thermostat with a flexible tube and a finned dry bulb is placed between the recirculated air bypass and the air reheater. This thermostat actuates a three way valve in the water lines leading to the air washer spray nozzles and to the finned tube water coil used for either heating or cooling air. By these means, dewpoint control of the air leaving the apparatus may be effected as the three way valve operates to deliver water of the proper temperature either to the water coil

[*Concluded on page 230*]



Details of Experimental Air Conditioning Installation at University of Illinois

Ducts

The circular duct sections are No. 14 U.S. ga galvanized sheet steel welded together. All other duct sections are No. 16 U.S. ga or heavier galvanized sheet steel. The heavy material was chosen so that stiffeners other than standing seams would not be necessary. For the most part the outside air, the recirculating air and the discharge air ducts are self supporting. Special effort was made to have the duct work air tight.

Air Filters

The air filters are of the dry type having ten pockets fitted with sheets of cellulose material. The air filtering surface totals 70.4 sq ft.

Atomizing Mist Nozzles

In order to regulate the humidity of the air entering the apparatus at any season, six atomizing mist nozzles are located between the air preheater and the chilled water air cooling coil. The nozzles are in a single bank and their discharge is directed against the air stream. The nozzles are under the control of a pneumatic, adjustable, duct hygrostat which operates a reverse acting modulating valve in the water line leading to the nozzles. A fine mesh copper wire screen between the mist nozzle chamber and the air cooling coil removes unvaporized drops of water from the air.

Air Heating and Cooling Coils

These are all of the extended surface type having horizontal copper tubes with 85 thin aluminum fins per foot of tube length. Those coils which contain either steam or water have the copper tubes fastened into a vertical header at each side of the coil section. The finned tube, direct expansion air cooling coil has two liquid refrigerant headers each serving six of the twelve horizontal serpentine tube circuits and one suction header which serves all twelve refrigerant circuits. All coils were selected so that the air velocity will not exceed 500 fpm at the coil face area when the plant is handling 3000 cfm measured at 70 F and 29.92 in. of mercury. All coil faces are 24 in. high and 36 in. wide and have a face area of 6 sq ft.

The air preheater coil has sixteen ½ in. O.D. horizontal copper tubes placed in a single vertical row. The air reheater coil is similar to the preheater coil except that it has thirty-one ½ in. O.D. horizontal copper tubes arranged in two vertical rows with the tubes staggered in the rows. Each coil may be operated with any steam pressure up to 15 lb per sq in. ga.

The finned coil, which may be used with either hot or cold water, consists of four sections. These sections are duplicates of the air reheater coil and are so arranged that the water may be circulated through either one, two, three, or four sections connected in series. This allows the use of either two, four, six or eight vertical rows of tubes for studying heat transmission in such coils with forced circulation of either hot or cold water. Counterflow of the water and air occurs and the piping is so arranged that the water is fed into each section at the bottom on one side and is taken out at the top on the opposite side. These four coil sections have a common expansion tank which insures that they are always completely filled with water under all conditions of operation.

The direct expansion "Freon" air cooling coil has seventy-two ¾ in. O.D. horizontal copper tubes arranged in six vertical rows with twelve tubes in each row. The tubes are staggered in the vertical rows. The liquid refrigerant is supplied through a thermostatic expansion valve to the first vertical row of tubes by means of two vertical headers which have metering orifices to insure the equal distribution of the refrigerant to the tubes.

Each air cooling coil is placed over a drip pan with a drain connection to an insulated condensation receiver. The vapor condensed from the air passing over the coil surfaces may be either drained to the sewer or weighed in a tank placed on scales.

The specified capacity of the direct expansion air cooling coil and also of the four section chilled water air cooling coil, supplied with 50 gpm of chilled water entering at 45 F, is the equivalent of 12 tons of refrigeration when the air enters the coils with a relative humidity of 40 per cent and a dry bulb temperature of 85 F. This capacity is based on the plant handling the equivalent of 3000 cfm measured at 70 F and 29.92 in. of mercury.

Air Washer

The air washer has 20 spray nozzles arranged in two banks of 10 each with the discharge of the spray water in the direction of the air flow. Deflecting louvres are situated in the air stream at the inlet end of the washer. The air outlet end of the washer has 24 vertical eliminator plates which are kept flooded by nine additional spray nozzles. The water supply pump delivers 50 gpm to the washer nozzles at a pressure of 25 lb per sq in. ga. The water piping is arranged for the use of either recirculated, heated, or chilled spray water. The maximum air velocity in the spray chamber does not exceed 450 fpm when the plant is operating at full capacity.

Fan

The fan is a backward curved blade, single inlet, single width unit which delivers 3000 cfm at 1150 rpm under standard conditions with the present setup. The fan motor has been selected with sufficient capacity to allow the fan to operate to deliver 3000 cfm, at standard conditions, against a static pressure of 1.5 in. w. g. greater than the total resistance of the present installation. By means of manual control of the motor the fan may be operated at nine different speeds.

Compressor

The "Freon" compressor has a rating of 13.1 tons of refrigeration when operating at a speed of 450 rpm with 38 F suction temperature and a condensing pressure of 110 lb per sq in. ga. The two 5¾ in. by 4 in. single acting air cooled cylinders are representative of the latest type of design. The compressor motor is arranged for automatic starting and stopping and facilities are included for five different operating speeds by manual adjustment of the starter rheostat.

Condenser

The "Freon" condenser is of the shell and tube type with seven water passes and has a total of fifty-four 1¼ in. tubes. The diameter of the shell is 16 in. and the overall length is 9 ft 9 in. The condenser has a liquid pot welded to the underside of the shell which serves as a receiver thereby reducing the amount of "Freon" required in the system. The water supply to the condenser is under the control of a regulating valve in the water supply line which is actuated by the refrigerant pressure within the condenser shell. The condenser is also fitted with a water pressure failure switch which stops the compressor motor whenever the water supply is shut off.

Water Cooler

The "Freon" cooler, to provide chilled or refrigerated water, has three consecutive horizontal water passes arranged one above the other. Each water pass consists of two 2⅛ in. O. D. copper tubes 18 ft 6 in. long. The tubes of each pass are connected in parallel by means of headers at each end. Within each 2⅛ in. copper tube are five ⅝ in. O. D. copper tubes through which the water flows. Each water pass is fitted with a thermostatic expansion valve which supplies liquid refrigerant to one end of each of the 2⅛ in. O. D. copper tubes. The expanded refrigerant flows in a single direction around the small water tubes and is removed from the water passes by three connections to a common suction header. Suitable header arrangements at each end of the 2⅛ in. O. D. copper tubes allow the proper flow of refrigerant and water. The refrigerant pressure in the water cooler is controlled by an adjustable back pressure valve in the suction line leading to the compressor. The back pressure valve functions to maintain a predetermined pressure of the refrigerant in the evaporator sections of the water cooler and is a protective device.

Compressor, Condenser and Water Cooler Controls

In connection with the condenser and the compressor suction line are three pressure actuated electrical switches. These are a high pressure cutout, which opens when the condenser pressure becomes excessive; a low pressure cutout, which functions when the pressure in the suction line drops below a desired value; and a water pressure failure switch which opens when the water supply to the condenser is stopped. The three electrical switches are wired in series and the compressor motor cannot be automatically started until they are all closed. Anything which causes one of the three switches to open will cause the compressor motor to stop.

The water cooler is protected against freezing by a thermostatically operated switch in an electrical circuit to a solenoid valve in the liquid refrigerant line between the condenser and the water cooler. When the water temperature falls to that for which the thermostat is adjusted the electrical circuit is broken, the solenoid valve closes and the compressor pumps down its suction line until the low pressure cutout causes the compressor motor to stop. The thermostatic bulb of the low temperature control is placed in the line leading away from the water cooler outlet.

The solenoid valve in the liquid refrigerant line is also under control of a hand operated switch at the fan motor control panel and a pressure switch actuated by a thermostat placed in the main air duct just ahead of the reheater coil. In order to secure continuous operation of the compressor these latter two switches and the thermostatically operated switch at the water cooler outlet must be closed, as all are wired in series. The opening of any one of the switches will cause the solenoid valve in the liquid line to close and the low pressure cutout in the suction line will stop the compressor.

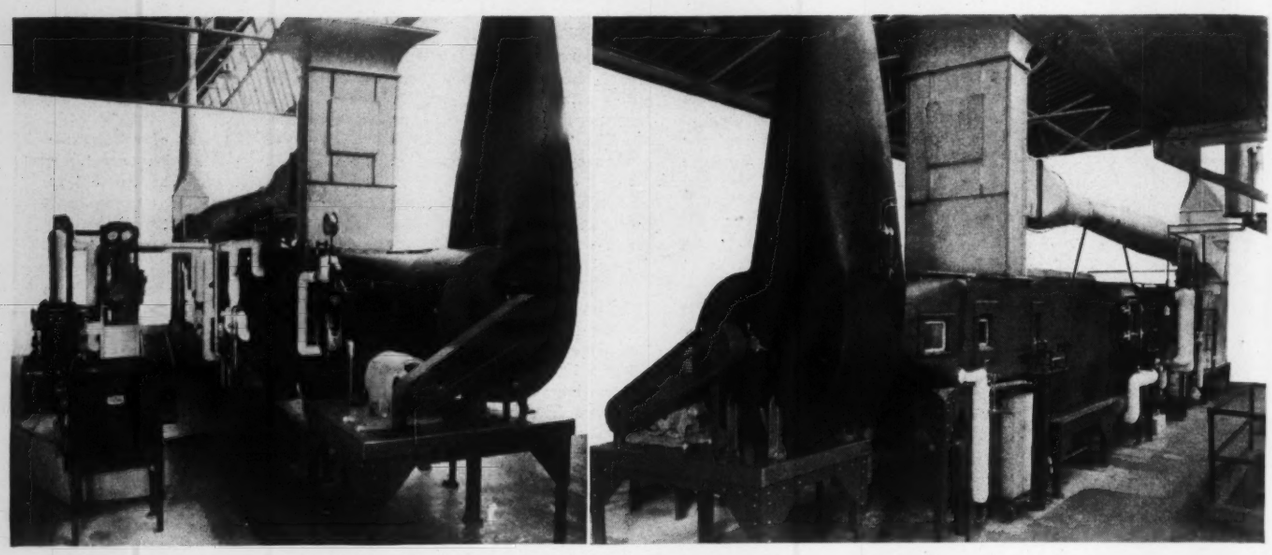
Water Pump and Heater

The centrifugal pump is a single stage, single inlet unit direct connected to a 3 hp motor. The pump has a capacity of 50 gpm against a discharge pressure of 25 lb per sq in. ga.

The water heater is a shell and tube type heat exchanger having two water passes. The capacity of the heater is 600,000 Btu per hr when heating water with 5 lb steam.

Motors

All motors are 230 volt direct current machines operating at 1750 rpm at full load. The direct connected motor for the water pump is rated at 3 bhp. Both the fan and the compressor are driven by multiple V belts. The fan motor is a 3 bhp unit and the compressor motor is a 15 bhp machine. No automatic starting and stopping features are incorporated in the pump and fan motor controls. All motor starting controls have a no-voltage release and the compressor motor may be operated by hand when any or all protective devices function to stop the machine.



Left—South side of equipment, showing compressor, condenser, water cooler and main assembly of air handling units. *Right*—View of north side showing observation windows and access doors in main assembly, and scales and tanks for weighing condensate

or the air washer. This thermostat also operates the electrical switch in the circuit of the solenoid valve in the "Freon" liquid line when the direct expansion air cooling coil is used. Whenever necessary or desirable this electrical switch can be rendered inoperative by the use of a stop and waste cock in the compressed air line leading to it.

Two additional gradual acting duct thermostats are installed, one in the recirculated air duct and one in the discharge air duct near the fan outlet. A three way cock is located in the air supply line leading to the two thermostats just mentioned. Dependent upon the position of the plug in the three way cock, the air reheater coil may be controlled either by hand operation of a steam valve, by the action of the thermostat at the fan outlet on the reheater coil modulating valve, or by the action of the recirculating duct thermostat and the fan discharge thermostat working in series to manipulate the modulating steam valve of the reheater coil. These arrangements permit regulation of the air temperature as it finally leaves the apparatus.

Problems for Which Apparatus Is Suitable

The apparatus is suitable for use in the study of many problems including: (1) Air cleaning by use of either filters or washers. (2) Air humidification. (3) Air cooling and dehumidification by three different pieces of apparatus. (4) Hot blast heating using either steam or hot water as a heating medium. (5) Heat transfer of finned coils using either steam or hot water as a heating medium. (6) Heat transfer of finned tube cooling coils, using either chilled water or direct expansion of the refrigerant, with dry and wet surfaces. (7) Problems involving the reheating of cooled and dehumidified air by use of either a steam reheating coil or by bypassing recirculated air. (8) All-year air conditioning of spaces in which typical load conditions may be maintained either in summer or winter. (9) Centrifugal fan performance under different load conditions. (10) Precision measurements of both dry and wet bulb air temperatures. (11) Air distribution by means of nozzles, grilles, diffusers, etc. (12) The measurement of air flow and friction losses in ducts of varying section and shape and different units such as filters, coils, etc.

Germicide in Spray Water Reduces Bacteria in Air

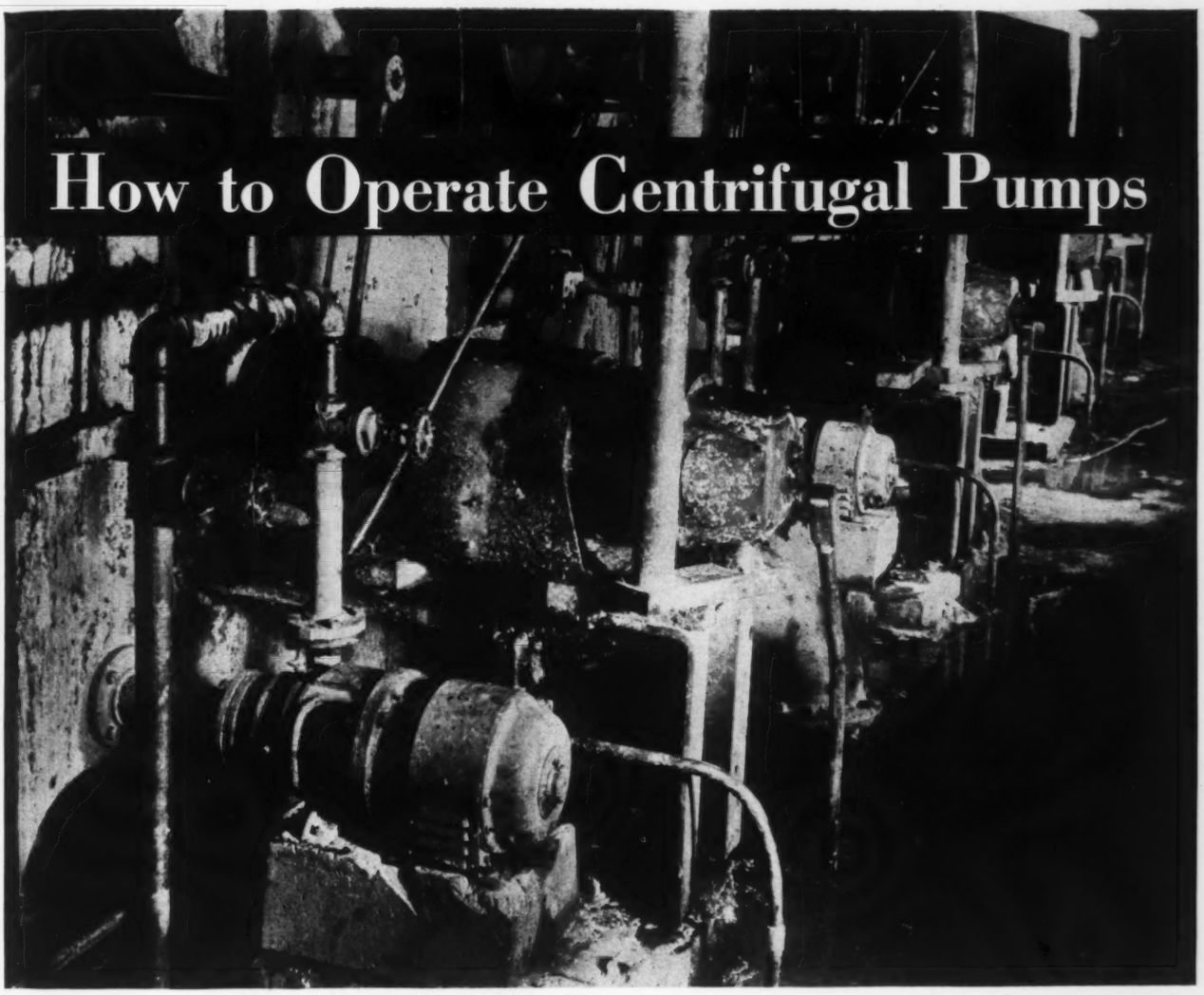
AIR washing systems offer an excellent opportunity for purifying air of bacteria as well as dust and pollen, according to T. S. Carswell, J. A. Doubly, and H. K. Nason who described experiments on a system for a general office in a paper presented before the *American Chemical Society* recently which has been published in the January issue of *Industrial and Engineering Chemistry*. In some cases bacterial content of air washed with water treated with a germicide was less than one-sixth that of air washed with untreated water. Recent studies by Wells and others at Harvard show that bacteria in minute liquid droplets do not settle but may drift for hours on currents of air.

Most of the common antiseptics cannot be used for this purpose as they impart an odor to the air, and because the amount of air in contact with the spray water is enormous, a substance with a distinctive odor may seriously taint the air even though its vapor pressure is very low. Mercurials, copper salts, and ammonium salts are unsatisfactory because they are corrosive.

The ideal germicide was found to be a commercial mixture of σ- and ρ- benzyl phenols, which met the requirements of being highly toxic to the organisms, noncorrosive, odorless in the required concentrations, nontoxic to man and the higher animals, stable even on prolonged aeration, easily dissolved in water, and economical and safe to use. These benzyl phenols also have the advantage of being toxic to algae and are therefore adapted to slime control.

No conclusions as to the possible hygienic significance of the treatment are made or inferred, the report being limited to physical measurements. The authors state that the possibilities of such treatment in process air conditioning in the fermentation industries (breweries, bakeries, etc.) for the elimination of mold and bacteria are interesting independent of the health aspect of purifying air by this means.

How to Operate Centrifugal Pumps



THE new "Standards" issued by the *Hydraulic Institute* include information on construction, selection, installation and operation of various types of pumps. The following instructions for proper operation of centrifugal pumps have been taken from this publication with the permission of the *Institute*.

Priming the Pump

Before starting the pump the casing and suction line must be filled with water. The pump must not be run unless it is completely filled with water, as there is danger of injuring some of the parts of the pump which depend upon water for their lubrication. Wearing rings will not seize when the pump is filled with water but are very liable to do so when the pump is run dry.

The pump may be primed by any of the following methods, as may be best suited to the conditions: priming by ejector: or exhauster: when steam, high pressure water, or compressed air is available, the pump may be primed by attaching an air ejector to the highest points in the pump casing. This will remove the air from the pump and suction line provided a tight valve is installed in the discharge line close to the pump. As soon as the ejector waste pipe throws water continuously, the pump may be started. After starting, a steady stream of water from the waste pipe indicates that the pump is primed. If this stream of water is not obtained, the pump must be stopped at once and the process of priming repeated. A foot valve is unnecessary when this kind of device is used.

Priming by Foot Valve Only: When it is not practicable to prime by ejector or exhauster, the pump may be primed by the use of a foot valve. The foot valve will prevent water running out through the suction inlet and the pump can be completely filled with water from some outside source. Pet cocks on top of the pump should be opened during filling to allow the air to escape. A tight foot valve will keep the pump constantly primed, and thus it may be used for automatic pump operation. It must be inspected frequently, however, to see that it does not develop leaks and thus allow the pump to be started dry.

Priming by Vacuum Pumps: When neither of the above methods of priming are practicable, the pump may be primed by the use of a vacuum pump to exhaust the air from the pump casing and suction line. A wet vacuum pump is preferable, as it will not be injured if water enters it. When a dry vacuum pump is to be used, the installation must be such as to prevent water being taken into the air pump. This may be done by having

the pipe leading from the pump to the air pump installed with a vertical rise of over 35 ft.

Careful attention to the priming method when the pump is installed may save annoyance later due to improper priming equipment.

Starting and Stopping

The pump must not be started without being primed. Be sure that the driver rotates in the proper direction. This is usually indicated by a direction arrow on the pump casing.

A high or medium head centrifugal pump when primed and operated at full speed with the discharge gate valve closed requires much less power than when operating at its rated capacity and head with the discharge gate valve opened. For this reason, it is advantageous that the gate valve always be closed when the pump is being started, except as noted below. It is to be noted, however, that in general the lower the normal operating head, the less does the closing of the discharge gate valve affect the power required at normal capacity; until at extremely low heads such as for low head, large volume irrigation pumps, where the design approaches the mixed flow type of pump, the shut-off horsepower may equal or exceed the horsepower required with discharge valve open.

Pumps of the mixed flow type frequently require more horsepower with the discharge valve shut than with it open. Propeller type pumps nearly always require a great deal more power at shut-off than at rating and must be started with the discharge valve open. The manufacturer's instructions should be consulted for the characteristics of such pumps.

Reduced Voltage Starting: With smaller sizes of pumps with a squirrel cage induction motor it is often permissible to start the motor on reduced voltage with the gate valve open, if the torque conditions of the motor permit such starting and no likelihood of injurious water hammer will occur when changing the motor over from the starting position to the running position.

With larger sizes of pumps using squirrel cage induction motors of the reduced voltage starting type, it is recommended that the gate valve always be closed when the pump is being started, except as noted above.

Across the Line Starting: When using either small or large squirrel cage induction motors of the line starting type, it is permissible to have the gate valve open when the pump is being started. The length of time of the disturbance on the line, due to the starting condition may be shortened if the gate valve remains closed.

Synchronous Motors: Synchronous motors of the general purposes sizes up to and including 500 hp at 80 per cent power factor, and 600 hp at unity power factor, having speeds of 500 rpm or higher are designed with sufficient pull-in torque to start pumps with discharge valve open. Above these horsepower ratings and for speeds below 500 rpm, motors are not designed standard for sufficient pull-in torque to start pumps with discharge valve open. However, the comments given above for squirrel cage motors also apply to the synchronous type.

Wound Rotor and Direct Current Motors: If the pump must be started with the discharge valve open and the starting current must be kept to a minimum, a wound rotor induction motor should be used, as this type of motor will develop full load torque without requiring excessive line current; or, if direct current is available, a motor of the direct current type will also develop full load torque without taking line current exceeding 125 to 200 per cent of normal full load current.

Stuffing Boxes: Do not draw glands tight. A slight flow of water from the stuffing boxes when pump is running keeps packing in good condition.

Wearing Rings: Wearing rings are fitted in the casing and usually on the impeller around the inlet to reduce the quantity of water leaking back from the high pressure side to the suction side. These wearing rings are fitted with a small clearance and depend on the water in the pump for lubrication. They will eventually wear so that the clearance becomes greater and more water passes back into the suction. The rate of wear depends on the character of the water pumped.

Stopping: Before stopping the pump, close gate valve. This will prevent water hammer on check valve.

Locating Pump Trouble

No Water Delivered—
- (a) Pump not primed.
- (b) Speed too low*
- (c) Discharge head too high.
- (d) Suction lift higher than for which pump is designed.
- (e) Impeller completely plugged up.
- (f) Wrong direction of rotation.

Not Enough Water Delivered—
- (a) Air leaks in suction or stuffing boxes.
- (b) Speed too low.*
- (c) Discharge head higher than anticipated.
- (d) Suction lift too high. Check with gauges. Check for clogged suction line or screen.
- (e) Impeller partially plugged up.
- (f) Not enough suction head for hot water.
- (g) Mechanical defects: Wearing rings worn. Impeller damaged. Casing packing defective.
- (h) Foot valve too small.
- (i) Foot valve or suction opening not submerged deep enough.

Not Enough Pressure—
- (a) Speed too low.*
- (b) Air in water.
- (c) Mechanical defects: Wearing rings worn. Impeller damaged. Casing packing defective.
- (d) Impeller diameter too small.

Pump Works for a While and Then Loses Suction—
- (a) Leaky suction line.
- (b) Waterseal plugged.
- (c) Suction lift too high.
- (d) Air or gases in liquid.

Pump Takes Too Much Power—
- (a) Speed too high.
- (b) Head lower than rating, pumps too much water.
- (c) Specific gravity or viscosity too high.
- (d) Mechanical defects: Shaft bent. Rotating element binds. Stuffing boxes too tight. Wearing rings worn. Casing packing defective.

*When direct connected to electric motors, check up whether motor is across the line and receives full voltage. When direct connected to steam turbines, make sure that turbine receives full steam pressure.

E. Vernon Hill Offers a Solution to*

The Problem of Condenser Water for Large Air Conditioning Plants

AS air conditioning installations increase in number and size certain problems are presented, some of which are physiological, some mechanical, and some economic. The problem of condenser water is one of the greatest interest in many localities where the water rate is high or where the supply is inadequate. It is pressing for immediate solution in Chicago, where the water rate is the lowest of any large city in the world, but for a different reason.

Loran D. Gayton, the city engineer of Chicago, recently read a paper before the Illinois chapter of the A.S.H.V.E. on air conditioning in the central business district of Chicago and its relation to the water supply and the sewers, which was an exceptionally valuable and comprehensive treatment of the subject.[1] Mr. Gayton discussed at some length the air conditioning load in the loop district and the probable rate of increase in the immediate future. He explained how water was supplied to this district, in what quantities, and to what extent it could be increased to meet the increasing demand.

He then took up the more important part of this local problem, the sewer system in the downtown district, presenting in some detail the history of sewer construction in Chicago. He pointed out that the present sewer system was designed by one of the great engineers of this country, Ellis S. Chesbrough, and installed between 1855 and 1860. This was 77 years ago and the sewage system was designed to take care of a normal increase in population for the succeeding 25 years. It was designed, furthermore, at a period when there were no skyscrapers, the pavements on the streets were not impervious to water, and there was a considerable area of private land not roofed or paved.

Not only has the condition changed markedly since this sewage system was installed, but the system has been obsolete for 50 years. The result is that the sewers are overloaded at the present time, even without the enormous addition of condenser water from cooling plants; basements are frequently flooded and considerable damage done because it is impossible for them to meet present day demands, not only after severe storms but in hot, dry weather when the air cooling load is at its maximum.

The problem that confronts Chicago, therefore, is not what is going to be done in the future about condenser water but what is going to be done now.

The Central District Water Supply

Fig. 1 represents the water supply in the downtown district, bounded on the north and west by the Chicago river, on the east by Michigan Ave., and on the south by Roosevelt Road. Into this district we find at the north boundary three large mains for supplying water from the Chicago Ave. pumping station; one a 24 in. main, passing under the river at Michigan Ave.; another 24 in. main at LaSalle St., and also a 36 in. main. These three mains have a combined capacity of 40,000,000 gal of water per day at ordinary velocities. The average velocity is usually considered as 5 ft per sec.

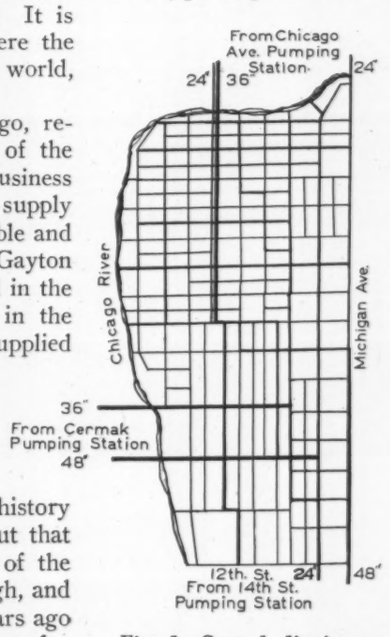


Fig. 1—Central district water supply system of Chicago

Passing down the river boundary on the west we find two mains entering from Cermak pumping station, one at Harrison St. of 36 in. diameter, and one at Polk St. 48 in. in diameter. These two mains have a combined capacity of 65,000,000 gal per day.

On the south boundary of the loop are two more mains, from the 14th St. pumping station, one 48 in. in diameter entering the loop at Michigan Ave. and one 24 in. in diameter at Wabash Ave. These two mains have a combined capacity of 55,000,000 gal per day.

The total available water supply, therefore, to this district, at ordinary pressure (28 lb) is 160,000,000 gal per day. These figures are from Mr. Gayton's report. He further states that in case of fire, or any other emergency, these feeder mains can be worked to at least 50 per cent greater capacity, making the possible delivery of water 240,000,000 gal per day.

Most people have the idea that the flow of water in a street main is a more or less regular and definite proposition; that it flows in a certain direction at a rather high velocity. Sometimes this is true, often it is not. The city engineers can usually determine from their tests and official data what the average pressure should be in any given locality, also the direction and velocity of flow. But when we consider that such a water supply system is in reality an enormous reservoir or grid of inter-connected mains and branches, it can readily be appreciated that the velocity and even the direction of flow varies materially under different conditions. The flow may be southward at 5 ft per sec, let us say, in an 8 in. main on State St.; somewhere along the main a

*Member of Board of Consulting and Contributing Editors.

[1]See HEATING, PIPING AND AIR CONDITIONING, November, 1936, pp. 609-612.

connection is made that withdraws a considerable amount of water; this flow may be sufficient to reduce the pressure materially below this point and even cause a reversal of flow so that the branch connections on intersecting streets south of the point will have to supply water for this particular main.

In general the large mains show a consistent directional flow but in the smaller mains the behavior of the water passing through may be difficult to predict. It is perhaps best, therefore, to consider the water supply system of the loop district as a reservoir, in the form of a grid, with the supply mains previously mentioned as feeders. The thousands of connections to the mains and branches draw off the water in varying amounts during different hours in the day and different seasons in the year, the approximate average maximum being about 65,000,000 gal per day.

There are mains in practically every street, varying in size from as small as 4 in. up to the largest, which is 36 in. in diameter. In some streets there are two or three mains that have been installed from time to time to meet various load conditions and most of them cross connect at street intersections so that a substantially uniform pressure is secured throughout the district served.

One of the problems presented by the increasing number and capacity of air conditioning systems is that of condenser water—a problem of importance where water rates are high or the supply is inadequate, but pressing for immediate solution in Chicago, where rates are low and the supply ample, because of limited facilities for disposal. . . . Dr. Hill analyzes the situation and proposes his solution—to take water from the supply main, pass it through the condenser, put it back in the main. The service would be charged for on the basis of the amount of heat imparted to the water so used. At first thought, there are objections to the scheme; the author states them, then refutes them. . . . While this discussion is primarily of local interest, its application is broadened because Chicago is very much of an "air conditioned" city and a similar problem is now or will be faced in other cities.

The Sewer System of This District

Looking at the diagrammatic drawing of the water supply system of the district we can visualize about what the sewer system would look like if put on paper.

The district is served by main sewers in all of the east and west streets from Randolph St. south. These sewers are 3 ft in diameter at the outlets where they empty into the south branch of the Chicago river and are 2 ft in diameter at the summits, which are located between State St. and Wabash Ave. From the summits these sewers slope to the east to a main sewer at Michigan Ave., which is 4 ft in diameter at the outlet at the main river, and 2½ ft in diameter at the summit near Congress St., then sloping south to the intercepting sewer at 12th St. There are brick sewers 2 ft in diameter in the north and south streets extending from Randolph St. north to the main river. South of Randolph St. in the north and south streets there are tile pipe sewers 12 in. in diameter that empty into the brick sewers in the east and west streets, having a summit in the middle of each block.

The above description of the Chicago sewer system, which was quoted by Mr. Gayton from a report written in 1911, is the sewage system in use at the present time except that the mains now empty into the intercepting sewers of the Sanitary District of Chicago instead of the Chicago river. Obsolete and inadequate as this sewage system is it does not appear wise to condemn it and to construct a new system until a final disposition has been made of the question of subways.

The sewage system, as previously stated, is inadequate to take care of normal waste at the present time, to say nothing of the ever increasing flow from condensers used in cooling installations in the large office buildings, department stores, and hotels.

Referring again to Mr. Gayton's paper, we find that the total maximum demand for water in the loop district amounts to about 65,000,000 gal per day, which was the amount determined by a test on July 26, 1936, in dry, hot weather. The test was made on the feeder mains supplying this area shown in the map of the water system, Fig. 1.

There were, on September 1, 1936, 398 air conditioning installations in this territory with a total refrigerating capacity of 11,865 tons. Figuring 2 gpm of water per ton of refrigeration we find that the maximum required for condenser water is 34,000,000 gal of water per day. This figure has been verified by several checks on the water flow. In other words, the district requires 65,000,000 gal of water per day in the hot weather of which one-half, in round figures, is used for cooling the gas in refrigerating condensers.

There is provided, therefore, operated and maintained pumping stations that represent an investment of many millions of dollars, together with a vast net work of distributing mains and branches, and pumped 65,000,000 gal of water per day, one-half of which is not required as *water* but as a *cooling medium* for air conditioning installations. It is not utilized as water; it is not altered in quantity or quality; but it is thrown into the sewer system after being warmed a few degrees.

This is a sad reflection on our engineering intelligence. More than this, it is an economic crime. We not only waste half of the water supply, purified and distributed at enormous expense, but we dump it into the sewers which are already taxed beyond their capacity. And now we contemplate building a new sewer system to care for this unnecessary load.

A Solution of the Problem

The solution of this problem is quite obvious; it is my belief it would have been adopted long ago were it not for the general ignorance as to just why condenser water is necessary and how it is used, and to the deep seated prejudice against any so-called "tampering" with the city's water supply. Incidentally this word "tampering" is a term of devastating import. It is quickly seized upon

by objectors to any new method or idea, and any modification or improvement in things as they always have been need only be labeled "tampering" to raise a storm of illogical protest.

The procedure I suggest to solve this condenser water problem is to use the water from the city mains in the quantities required for cooling the gases used in refrigeration and return it *back into the main* from which it was taken. A radical step, to be sure, on first thought but a method which I believe to be thoroughly sound.

Fig. 2 is a sketch illustrating the simplicity of this scheme. Water is withdrawn from the main under its normal pressure of about 28 lb, passes through the building wall and through the condenser, and then is pumped back into the main. The water has not been altered in any way, it has not been contaminated or changed in volume; its temperature, however, has been increased about 10 deg.

The first objection that might be offered is that the water would be so warm it would not be palatable to drink. But when this criticism is analyzed it is found to have little weight. The water in the mains varies in temperature from spring to midsummer from about 40 F to its maximum of 75 F, and during the hot weather it is usually too warm to be palatable at any time, so it is invariably iced or refrigerated for this purpose.

If we assume that there are 65,000,000 gal of water coming into the loop daily, and the full 50 per cent is being used for condenser purposes, raised 10 deg in temperature and returned to the main, the total increase in the temperature of all the water would be only about 5 deg which would not ordinarily be detected by those who drink it without its being cooled. It is so much less than the temperature variation from month to month throughout the summer that the effect would be unnoticed. But who drinks water in downtown Chicago anyway? Very few people as I know them. Let us assume, however, that one-half of the 400,000 habitués and visitors to the loop district each day drink a pint of water each; this is a high estimate but it will do for our purpose. Two hundred thousand pints of water drunk daily by the people in the loop would be only 25,000 gal or less than 1 per cent of the total used. How would increasing the temperature 5 deg of the other 99+ per cent affect its use?

It is of course obvious that in hotels where large amounts are used for cooking, for tubs and shower baths, and for cleansing purposes, flushing toilets, etc., in the laundries, an increase in temperature would result in a definite saving as the water must be heated. In sprinkling the streets, for tank water in office buildings, for sprinkler systems, and sanitary uses, an increase in the temperature certainly could do no harm. On the whole it is probably true that a distinct saving would result to the users of water in the district if it were warmed slightly by use in refrigerating condensers.

The next criticism might be that if the water piping in the condenser should spring a leak refrigerating gases would enter the water main. This, of course, is true but accidents of this kind are very exceptional indeed and no harm could result in rare instances. In CO_2 installations a leakage of carbon dioxide gas into the water main could cause no possible harm. The water might be a little more sparkling and pleasant perhaps than otherwise, but it would be a very large cooling installation where the amount of CO_2 that entered the main in this way could be detected. The same is true in condensers using "Freon," which, on entering the main, would be immediately dissipated as a gas; it is harmless and nonpoisonous and could have no deleterious effect.

Furthermore, if a leak in the condenser did occur the leak alarm required by law in every installation would immediately notify the engineer in charge. Automatic valves could be installed to close the water supply line from the condenser immediately if such a leak occurred.

And if the objector is still not convinced and wishes to prevent any possibility of "Freon" entering the water mains a reciprocating pump could easily be installed between the condenser and the connection into the main and a water pressure of, say, 150 lb maintained in the condenser, in which case a leak would cause water to flow into the refrigerant lines instead of the refrigerant entering the water piping. A suitable trap of course would have to be provided in the liquid line leaving the condenser so that the water would not enter and damage the expansion coils.

The third objection, which is the only other one I have heard advanced, is the danger of some ignorant or unscrupulous person tampering with the condenser water line and in some way getting a cross connection with sewer or other waste piping, thus contaminating the water system. It is difficult to see how this could possibly occur any more than it could with any water connection to the mains. Even so it could be prevented without much difficulty if a rigid sys-

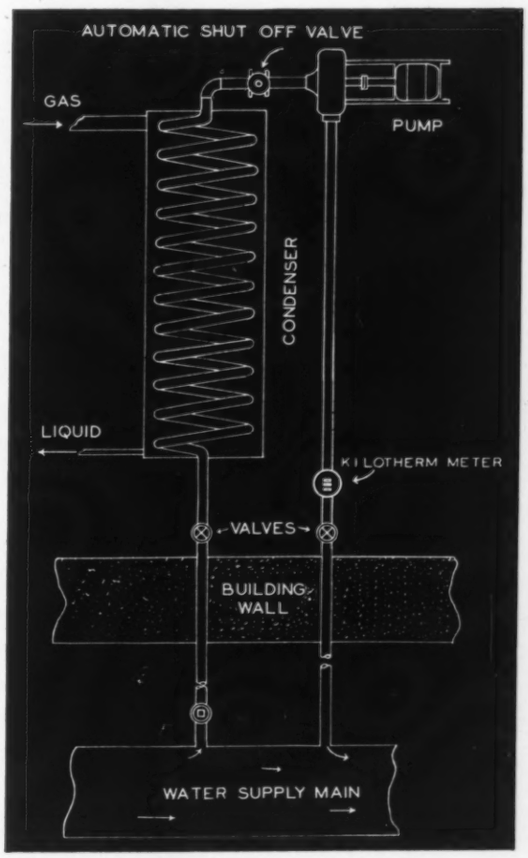


Fig. 2—Detail of suggested condenser water connection to city main

tem of permits and inspections by proper city authorities was required before any connections of this kind were allowed.

Benefits to Be Derived

Let us assume that prejudice can be overcome and the city fathers induced to give serious consideration to this solution of the condenser water problem; an ordinance would be drawn, forbidding the waste of condenser water to the sewers in installations larger, let us say, than 10 tons capacity. A permit would be secured from the city engineering department to make the necessary connections for all installations exceeding 10 tons in capacity. The permit would be for a new service, a service in which a different kind of meter would be substituted for the present water meter; it would be a meter calibrated in pounds of water and temperature rise and the product of the two would give the number of Btu of heat added to the water. Every thousand, or hundred thousand, Btu would be a therm (perhaps *kilotherm* would be the appropriate term for it), and the user would pay for the kilotherms used and not for water which he does not use.

The permit would be issued only after a survey by competent engineers from the water pipe extension department or the bureau of engineering, and a limit set, probably 10 deg, for the temperature rise allowed.

The method and all material used for the installation would be from specifications approved by city authorities and of course rigid inspection during installation would be required.

The charge for cooling service on the kilotherm basis should be equal to or somewhat in excess of the present rate for water, as the price of water in Chicago is very low compared with other cities throughout the country.

With this type of service for cooling, the amount of water supplied to the loop district would not have to be increased for many years to come; probably it could be materially reduced. Approximately one-half the load would be taken from the sewer system, which is the important end sought. It goes without saying that this reduction in the amount of water used would result in a very great saving to the city in supplying water which is now wasted, and the returns from the thermal service would be far greater than from the water sold as water at present.

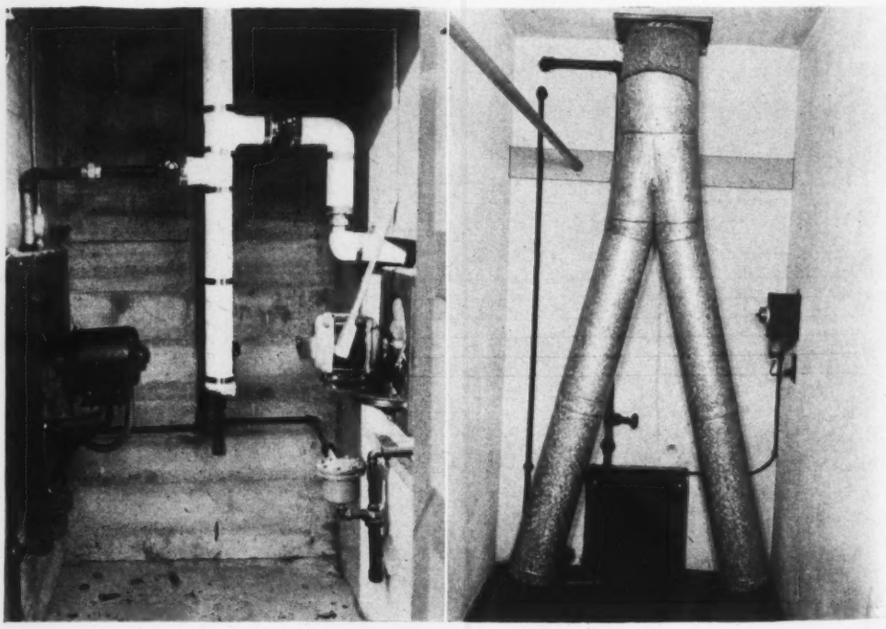
Unit Heaters Speed Drying of Cleaned Garments

UNIT heaters are 50 per cent faster than the coil system previously used for drying solvent cleaned and laundered garments, according to Chester Foreman, proprietor of Foreman's cleaning plant in Salem, Ore. Air in the room where garments are hung to dry and deodorize should be changed every 30 seconds if the temperature, approximately 130 degrees, is maintained, he says.

To accomplish this he has placed heaters in each of the deodorizing rooms some 30 in. above the floor. A heater in the long and narrow drying room for laundered garments was placed practically at floor level. In the corners of both deodorizing and drying rooms, and located on the same wall as the heaters, vent pipes 10 in. in diameter were installed 8 in. above the floor. These corner vent pipes unite in a larger exhaust pipe near the ceiling of the room, which is carried through to a pivoted hood on the roof.

Drying and deodorizing rooms are usually small and often narrow. Hot air driven from the heater strikes the floor some distance from the unit, bounds ceilingward, becomes saturated with moisture or solvent and descends at the end of the room containing the heater—or more particularly at the end corners where the venting pipes are located.

Three drying room heaters installed according to this principle have given Foreman's "absolute satisfaction." As compared with the coil system of drying, Mr. Foreman has found unit heaters gratifyingly economical in respect to steam consumption.—BEN MAXWELL.



***Left*—The unit heaters for the drying and deodorizing rooms are in a room apart and are operated by explosion proof motors. *Right*—Vent pipes are at the same end of the room as is the unit heater**

Performance of Heaters and Coolers Under Changed Operating Conditions

By Bayard P. Fonda*

Not only must the engineer select the size of an air heating or cooling coil or the amount of heat exchange surface to meet given operating conditions, but he must also predict performance under a variety of changed operating conditions. . . . William Goodman (in his articles on solving heat transfer problems published in January, February and March) has explained how this may be done. Through a fortunate coincidence, Mr. Fonda submitted the present article at about the same time Mr. Goodman submitted his, so we have here another interesting equation and method for handling the difficult (as a rule) counterflow heat transfer problems

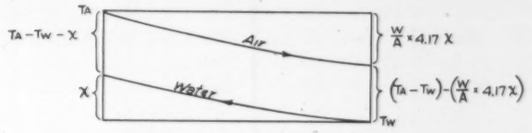
IT is a fairly simple operation to arrive at the size of an air heating or cooling coil or the amount of surface required to meet a given operating condition. However, the engineer must also predict the performance of cooling coils with inlet air and water temperatures or quantities quite different from those for which the coils were selected.

The solution of such problems is our subject here. I would like to emphasize right at the start that regardless of whether a process is one of heating or cooling, or whether the media are liquids or gases, the same principles of heat transfer govern the heat exchange. To simplify the treatment of this subject, a typical example will be used where air is being cooled by water in one of the familiar extended surface coolers of the counterflow type. The application of the discussion to other heat transfer problems will be mentioned briefly later.

For example, a certain cooling coil has capacity to maintain a room in summertime at 80 F. The air enters the coil at 80 F and leaves at a sufficiently low temperature to offset the heat gain in the room. Now suppose the inlet air temperature to the coil should change to 85 F or the inlet water temperature to the coil should change 5 or 10 deg. What would be the performance or heat removal of the coil under these new conditions?

To determine this is not so simple as one might think, for three conditions must be satisfied simultaneously. The heat gain of the water must equal the heat loss of the air and this heat exchange must satisfy the constants of the coil with the new logarithmic temperature difference which exists under the changed conditions. Since the new outlet air and outlet water temperatures are unknown, it follows that the logarithmic temperature difference between water and air is also unknown and the problem cannot be solved directly.

A solution rests in setting up the heat balance in terms of the inlet water and inlet air temperatures, which are known, and in solving for the outlet temperatures, which are not known. A mathematical trick relieves the expression for the logarithmic temperature difference of most of its terror and the rest is arithmetic. The real value of this solution rests in the fact that it applies to any heat exchange device, regardless of change in operating conditions. The inlet temperatures and the quantities of both the heated and the cooled media may be varied, whether they be liquids or gases, and the solution is still applicable. The derivation follows:



TA = Inlet air temperature
TW = Inlet water temperature
W = Weight of water in lbs. per hour
A = Weight of air in lbs. per hour.
H = Heat in B.T.U. per hour
K = B.T.U./Sq. Ft. Surface/°M.T.D/Hour
S = Effective surface Sq. Ft
θ = Logarithmic mean temp. diff. = $\frac{G.T.D. - L.T.D.}{2.3 \log \frac{G.T.D.}{L.T.D.}}$
X = Water temp. rise

(1) $H = WX = A \cdot .24 \cdot$ Air temp. drop

(2) Air temp. drop $= \frac{WX}{A \cdot .24} = \frac{W}{A} \times 4.17\,X$

$$H = KS\theta = \frac{(T_A - T_W - X) - \left[(T_A - T_W) - \frac{W}{A} \times 4.17\,X\right]}{2.3 \log \frac{T_A - T_W - X}{T_A - T_W - \frac{W}{A} \times 4.17\,X}} = \frac{.435\,X\left(\frac{4.17W}{A} - 1\right)}{\log \frac{T_A - T_W - X}{T_A - T_W - \frac{W}{A} \times 4.17\,X}}$$

(3) $$H = \frac{.435\,KSX\left(\frac{4.17W}{A} - 1\right)}{\log \frac{T_A - T_W - X}{T_A - T_W - \frac{W}{A} \times 4.17\,X}}$$

Equating (1) and (3) $H = WX = KS\theta$

(4) $$WX = \frac{.435\,KSX\left(\frac{4.17W}{A} - 1\right)}{\log \frac{T_A - T_W - X}{T_A - T_W - \frac{W}{A} \times 4.17\,X}}$$

Transposing and cancelling X

$$\frac{T_A - T_W - X}{T_A - T_W - \frac{W}{A}\,4.17\,X} = \text{Log}^{-1}\left(\frac{.435\,KS\left(\frac{4.17W}{A} - 1\right)}{W}\right)$$

$$= \text{Log}^{-1}\left[KS\left(\frac{1.81}{A} - \frac{.435}{W}\right)\right] = \text{"N" Temporarily}$$

$$T_A - T_W - X = N\left(T_A - T_W - \frac{W}{A}\,4.17\,X\right)$$

$$-X = N(T_A - T_W) - N\left(\frac{W}{A}\,4.17\,X\right) - T_A + T_W$$

Transposing :-

$$N\left(\frac{W}{A}\,4.17\,X\right) - X = N(T_A - T_W) - T_A + T_W$$

$$X\left[N\left(\frac{W}{A}\,4.17\right) - 1\right] = N(T_A - T_W) - T_A + T_W$$

(5) So $$X = \frac{N(T_A - T_W) - T_A + T_W}{N\left(\frac{W}{A}\,4.17\right) - 1}$$

Substituting the value of N in (5)

(6) $$X = \frac{\text{Log}^{-1}\left[KS\left(\frac{1.81}{A} - \frac{.435}{W}\right)\right] \times (T_A - T_W) - T_A + T_W}{\text{Log}^{-1}\left[KS\left(\frac{1.81}{A} - \frac{.435}{W}\right)\right] \times \left(\frac{4.17\,W}{A}\right) - 1}$$

Although Equation 6 appears formidable, it is not nearly as bad as it looks and about two minutes of arithmetic gives the answer. In solving the expression for the

*The Bryant Heater Co., Cleveland, Ohio.

anti-log N, it is recommended that logarithmic tables of at least four, and preferably five, places be used.

As an example, assume a certain cooler has been designed or selected to cool 5210 lb per hr of air from 80 to 70 F with 2500 lb per hr of water entering at 60 F. This represents a heat removal of $5210 \times 10 \times 0.24 =$ 12,500 Btu per hr. The water temperature rise must be $12{,}500 \div 2500 = 5$ F. The greatest temperature difference is $80 - 65 = 15$, the least temperature difference is $70 - 60 = 10$, and the logarithmic mean temperature difference (θ) of 15 and 10 is 12.3. This at once gives us the KS or constant of this cooler as $H = KS\theta$, and KS is $12{,}500 \div 12.3 = 1015$.

Now let us suppose that the inlet water temperature drops to 50 F and all other factors remain the same. What will be the total heat extraction of the cooler and what will be the outlet air temperature? Substituting the proper values in Equation 6 we get:

$$X = \frac{\text{Log}^{-1}\left[1015\left(\frac{1.81}{5210} - \frac{0.435}{2500}\right)\right](80-50) - 80 + 50}{\text{Log}^{-1}\left[1015\left(\frac{1.81}{5210} - \frac{0.435}{2500}\right)\right]\left(\frac{4.17 \times 2500}{5210}\right) - 1}$$

X = Water temperature rise = 7.5 deg,
$H = WX$ = 2500 lb × 7.5 deg = 18,750 Btu per hr,
Air temp. drop = 2500 lb × 4.17 × 7.5 deg ÷ 5210 = 15 deg,
Outlet air temp. = 80 — 15 = 65 deg.

Of course the result is secured in the same manner if the inlet air temperature is varied also. Furthermore, the water and air quantities may be changed provided the correction in the value of KS is made to correspond to the new air or water velocities. The variation in this factor for various air or water velocities is usually furnished by the coil manufacturer.

Appearance of Negative Quantity in Anti-Log

A disturbing condition that may arise in the use of Equation 6 is the appearance of negative quantities in the expression for the anti-log N. This means simply that the water temperature rise is greater than the air temperature drop and we are therefore trying to subtract the greatest temperature difference from the least temperature difference, giving a negative quantity. However, as the logarithmic mean of any two numbers is independent of sign, we are privileged to reverse the two numbers so that we still have a positive expression. This also effects the derivation of the denominator of Equation 6 so a change must be made in this. To summarize, if the expression for the anti-log N is negative, the form of Equation 6 changes to:

$$X = \frac{\text{Log}^{-1}\left[KS\left(\frac{0.435}{W} - \frac{1.81}{A}\right)\right] \times (T_A - T_w) - T_A + T_w}{\text{Log}^{-1}\left[KS\left(\frac{0.435}{W} - \frac{1.81}{A}\right)\right] - \frac{4.17W}{A}} \quad \ldots\ldots\ldots\ldots [7]$$

To exemplify a condition where Equation 7 must be used instead of Equation 6, let us increase the air quantity in the problem just discussed from 5210 lb per hr to 12,000. In this case $\left(\frac{1.81}{A} - \frac{0.435}{W}\right)$ in Equation 6 is negative, so Equation 7 is used. The KS for this particular cooler increases to 1300 because of the higher air velocity with the increased air flow.

Substituting the proper values in Equation 7 we have:

$$X = \frac{\text{Log}^{-1}\left[1300\left(\frac{0.435}{2500} - \frac{1.81}{12{,}000}\right)\right](80-50) - 80 + 50}{\text{Log}^{-1}\left[1300\left(\frac{0.435}{2500} - \frac{1.81}{12{,}000}\right)\right] - \frac{4.17 \times 2500}{12{,}000}}$$

X = Water temperature rise = 10.6 deg,
$H = WX$ = 2500 lb × 10.6 deg = 26,450 Btu per hr,
Air temp. drop = 2500 lb × 4.17 × 10.6 ÷ 12,000 = 9.2 deg,
Outlet air temp. = 80 — 9.2 = 70.8 deg.

When Expression for Anti-Log Equals Zero

There is a certain proportion for the quantities of air and water which will make the expression for the anti-log equal zero. If $\left(\frac{1.81}{A} - \frac{0.435}{W}\right)$ equals zero, the air temperature drop equals the water temperature rise and in this case the logarithmic mean temperature difference is indeterminate and can no longer be solved. But if this condition exists, the inlet and outlet temperature differences are the same. Therefore the arithmetic mean applies instead, and of course this is equal to the temperature difference either at the inlet or the outlet end. So θ becomes $(T_A - T_W - X)$. The derivation and final equation are now greatly simplified. Thus: $H = WX = KS\theta$, and θ becomes $(T_A - T_W - X)$. So

$$WX = KS\ (T_A - T_w - X)$$
$$WX = KS\ (T_A - T_w) - KSX$$
$$WX + KSX = KS\ (T_A - T_w)$$
$$X(W + KS) = KS(T_A - T_w)$$

$$\text{and } X = \frac{KS(T_A - T_w)}{W + KS} \quad \ldots\ldots\ldots\ldots [8]$$

To demonstrate the application of Equation 8 let us revert to our original example and assume an air quantity that makes $\left(\frac{1.81}{A} - \frac{0.435}{W}\right)$ equal zero. 10,400 lb per hr of air will do this and at the air velocity that results, the particular cooler in question has a KS of 1260. Substituting this value in Equation 8:

$$X = \frac{1260(80-50)}{2500 + 1260}$$

X = Water temp. rise = 10.04 deg,
$H = WX$ = 2500 lb × 10.04 deg = 25,100 Btu per hr,
Air temp. drop = 10.04 deg (The same as X, of course).
Outlet air temp. = 80 — 10.04 = 69.96 deg.

To apply the same solution to heat exchange equipment using other gases or liquids it is necessary only to substitute the proper value for the ratio of the specific heats of the heated and cooled media in the expression referred to on the diagram as the air temperature drop, $(W/A) \times 4.17X$. In this case, 4.17 is the ratio of the specific heat of water to that of air.

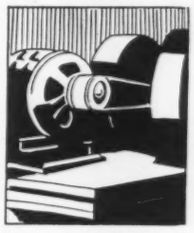
As these equations give the value of the temperature rise or drop of the liquid or gas and also the total heat exchange, it follows that the *efficiency* of a heat exchange device may be determined for a variety of inlet conditions.

The value of this solution in determining on paper the efficiency of boilers, furnaces, etc., for various operating conditions, is left to the reader.

Electricity Used by Air Conditioning

By William P. Rock*

HOW much electricity is used by air conditioning equipment? Information which is helpful in answering this question was obtained in a survey of monthly meter readings for the year 1936 of installations in Chicago which operated during the entire cooling season. The average demands and consumptions are given for the 15 classifications of business most interested, and, with a knowledge of the price of electric service in different localities, these figures may be readily translated into the cost of operation in cities with climatic conditions comparable to Chicago.



The installations treated here are all in the power company's residential and commercial classes of business, no industrial applications of air conditioning being included.

This survey shows the present average status of air conditioning in the principal classifications of establishments which have adopted it, and does not presume to show the ultimate status. As a picture of conditions existing in 1936, it necessarily includes several classes of business in which air conditioning has made great strides, but yet in which there are no completely air conditioned establishments in Chicago. Among these are hotels, none of which are completely air conditioned, also large offices and office buildings, some of which are yet air conditioned only in certain parts. On the other hand, the restaurant and theater classifications show a very complete picture because all of the many cases are fully air conditioned.

The meter readings of the individual customers were grouped to bring out the operating characteristics peculiär to the different classes of business. The readings of other meters showing the general use of light and power in the same establishments were also tabulated. In this way the total load and consumption of each establishment was obtained for comparison of air conditioning with the other uses of electricity.

Each of the 15 classifications analyzed is expressed in terms of the *average installation* for that classification. Some classifications are the result of averaging three or four establishments, and some are the result of averaging as many as fifty. Each is as inclusive as available records would permit. The charts and tables in a few cases do not show demand readings because in those classifications demand meters are not used.

To make the conclusions relatively adaptable to other cities, there is given a tabulation of the average weather conditions prevailing in Chicago month by month during the cooling season of 1936. While July showed a higher average temperature than August, August showed a greater usage of cooling equipment. This is attributable to the fact that the average humidity was higher in August than in July.

Some classifications start their cooling systems early in the season and others later. Some continue operation into November. Theaters, for instance, indicate little if any usage of air cooling after September.

The charts indicate the total electrical usage of the average air conditioned establishment in each classification. The shaded areas represent that part of the electrical usage caused by the air conditioning equipment. Where possible the chart for each classification shows both the kilowatthour consumption and the kilowatt demand for each month of the year.

The tabulation with each chart shows numerically the month by month kilowatthour and kilowatt readings on the air conditioning equipment only for the average establishment. The average kilowatthour usage per connected horsepower for air conditioning is also given for each classification. Since one horsepower operating one hour uses very close to one kilowatthour of energy, these figures indicate approximately how many hours, during the season, the average installation has been operated at the equivalent full load. For comparison the average usages per horsepower for the several classifications are grouped in a summary table.

[*See charts on pp.* 240-241]

Summary Table, Kwhr per Hp for Average Air Conditioned Establishment per Season

Establishment	Kwhr per Hp
Residential	
Room Cooler	514
Central System	769
Commercial	
Beauty Parlors	553
Doctors and Dentists	482
Funeral Parlors	310
Hotels	785
Offices, General	667
Restaurants	942
Stores	
Candy	911
Clothing	
Small	780
Medium	596
Department	505
Drug	1330
Shoe	675
Theaters	490

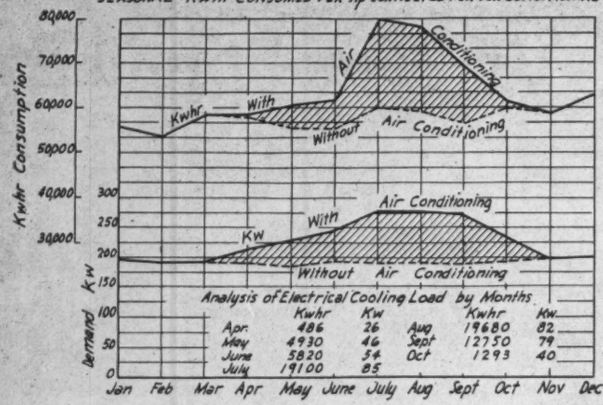
Weather Conditions, Chicago, March-November, 1936

Month	Average Temp. F	Relative Humidity, Per Cent		
		7 A. M.	Noon	7 P. M.
Mar	39.6	76	57	61
April	43.6	71	58	64
May	64.4	70	52	56
June	64.6	70	58	59
July	76.8	66	55	58
Aug	75.0	78	59	67
Sept	68.2	82	65	73
Oct	53.6	78	62	68
Nov	38.0	73	62	66

*Air Conditioning Division, Commonwealth Edison Co., Chicago, Ill.

GENERAL OFFICES

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 667

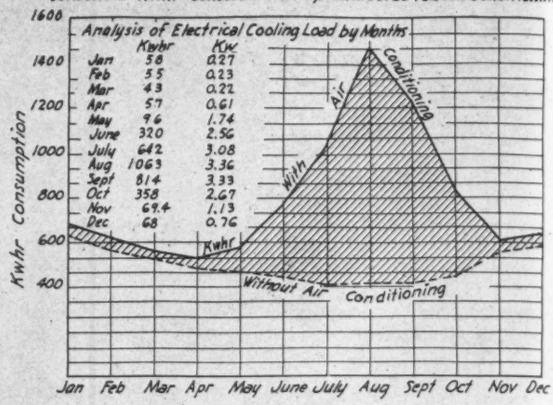


Analysis of Electrical Cooling Load by Months

	Kwhr	Kw		Kwhr	Kw
Apr	486	26	Aug	19680	82
May	4930	46	Sept	12750	79
June	5820	54	Oct	1293	40
July	19100	85			

RESIDENTIAL - CENTRAL

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 769

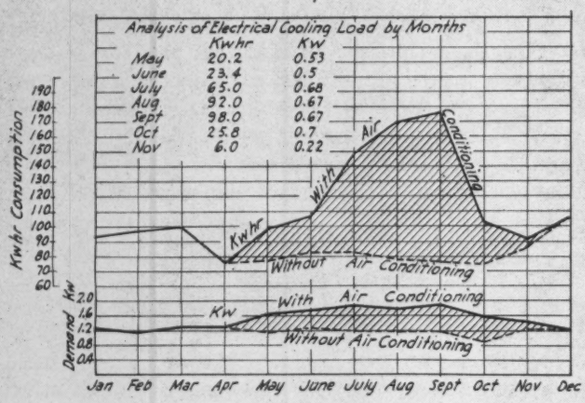


Analysis of Electrical Cooling Load by Months

	Kwhr	Kw
Jan	58	0.27
Feb	55	0.23
Mar	43	0.22
Apr	57	0.61
May	96	1.74
June	320	2.56
July	642	3.08
Aug	1063	3.36
Sept	814	3.33
Oct	358	2.67
Nov	69.4	1.13
Dec	68	0.76

DOCTORS' AND DENTISTS' OFFICES

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 482

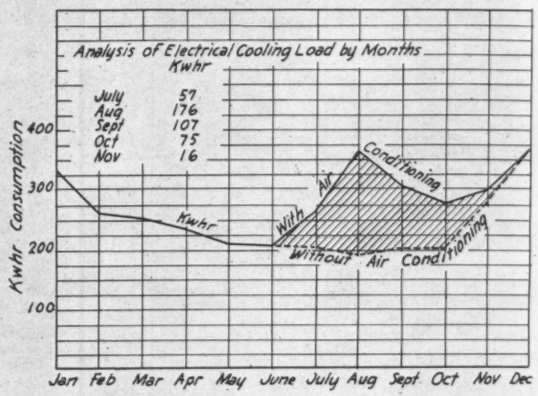


Analysis of Electrical Cooling Load by Months

	Kwhr	Kw
May	20.2	0.53
June	23.4	0.5
July	65.0	0.68
Aug	92.0	0.67
Sept	98.0	0.67
Oct	25.8	0.7
Nov	6.0	0.22

RESIDENTIAL - ROOM COOLER

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 514

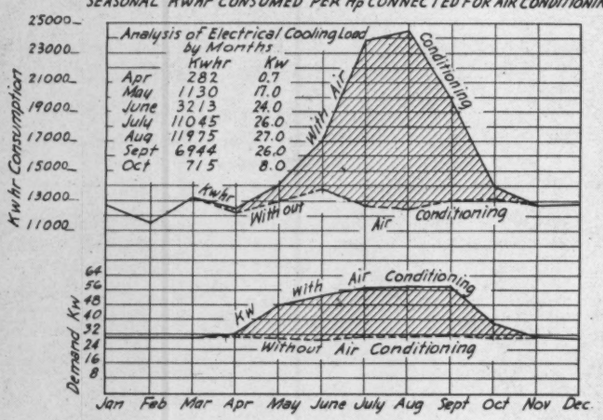


Analysis of Electrical Cooling Load by Months

	Kwhr
July	57
Aug	176
Sept	107
Oct	75
Nov	16

RESTAURANTS

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 942

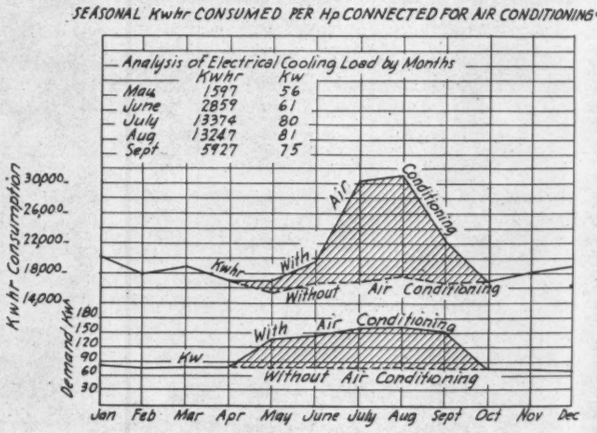


Analysis of Electrical Cooling Load by Months

	Kwhr	Kw
Apr	282	0.7
May	1130	17.0
June	3213	24.0
July	11045	26.0
Aug	11975	27.0
Sept	6944	26.0
Oct	715	8.0

THEATERS

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 490

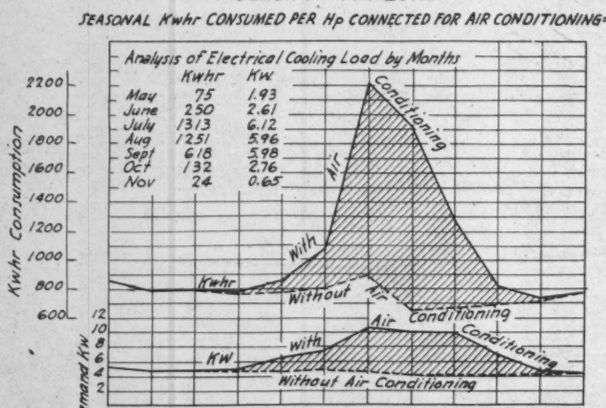


Analysis of Electrical Cooling Load by Months

	Kwhr	Kw
May	1597	56
June	2859	61
July	13374	80
Aug	13247	81
Sept	5927	75

BEAUTY PARLORS

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 553

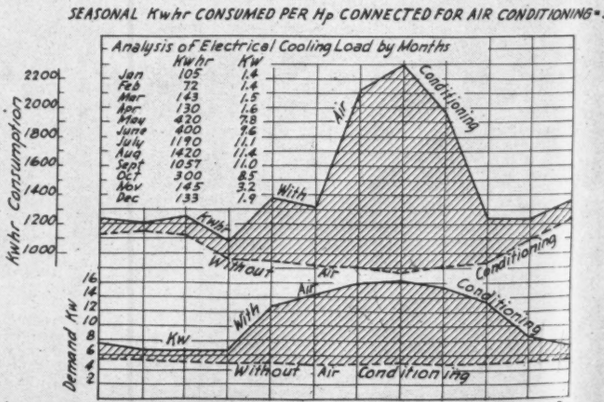


Analysis of Electrical Cooling Load by Months

	Kwhr	Kw
May	75	1.93
June	250	2.61
July	1313	6.12
Aug	1251	5.96
Sept	618	5.98
Oct	132	2.76
Nov	24	0.65

FUNERAL PARLORS

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 310

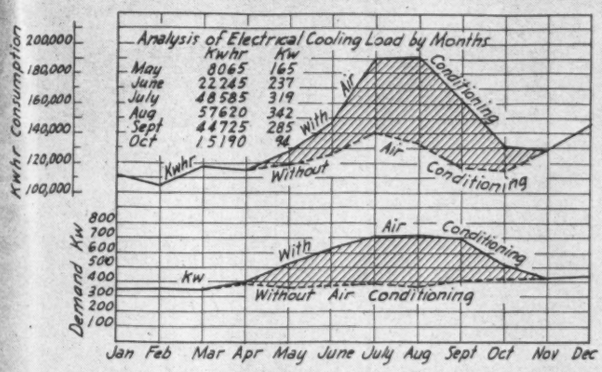


Analysis of Electrical Cooling Load by Months

	Kwhr	Kw
Jan	105	1.4
Feb	72	1.4
Mar	143	1.5
Apr	130	1.6
May	420	7.8
June	400	9.6
July	1190	11.1
Aug	1420	11.4
Sept	1057	11.0
Oct	300	8.5
Nov	145	3.2
Dec	133	1.9

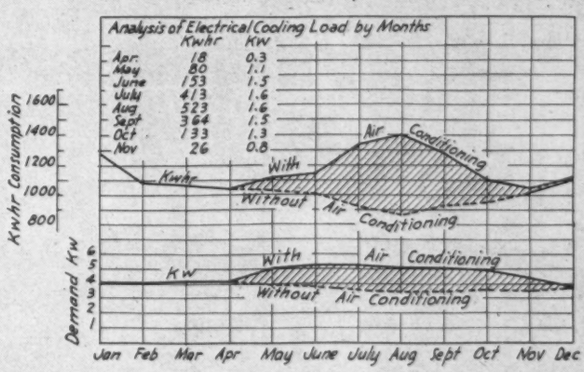
DEPARTMENT STORES

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 505



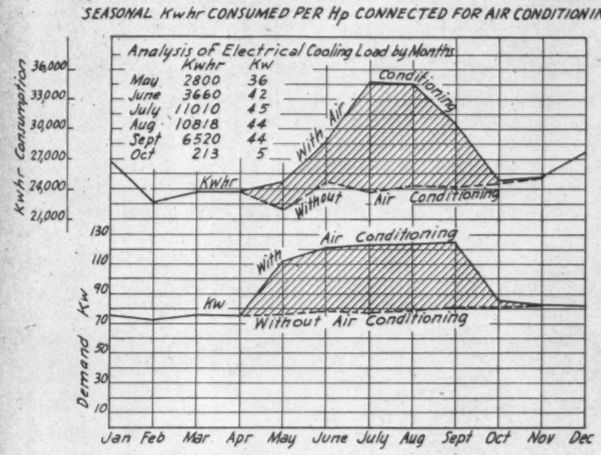
CANDY STORES

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 911



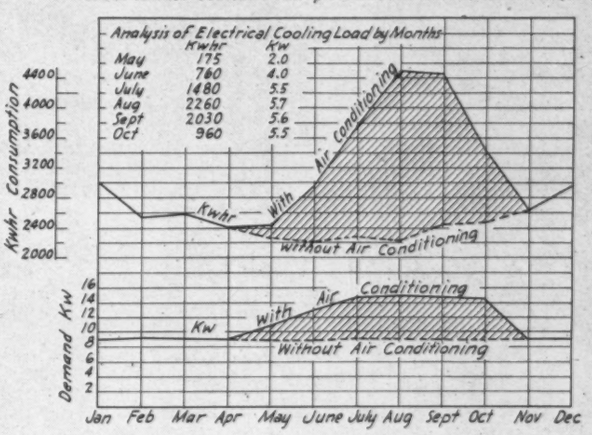
MEDIUM SIZED CLOTHING STORE

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 596



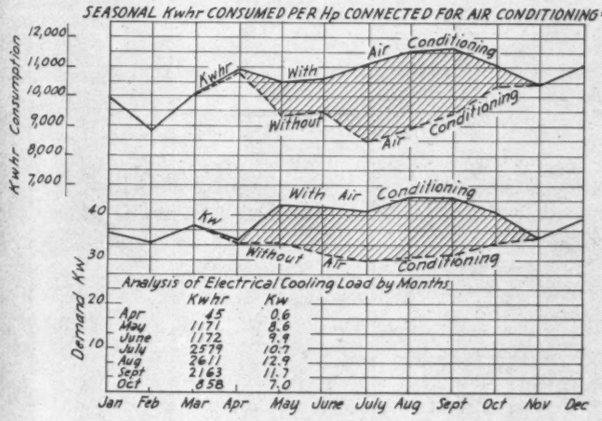
DRUG STORES

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 1330



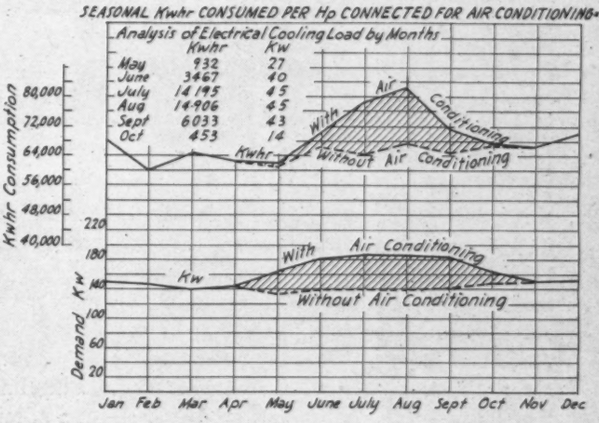
SMALL CLOTHING STORES

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 780



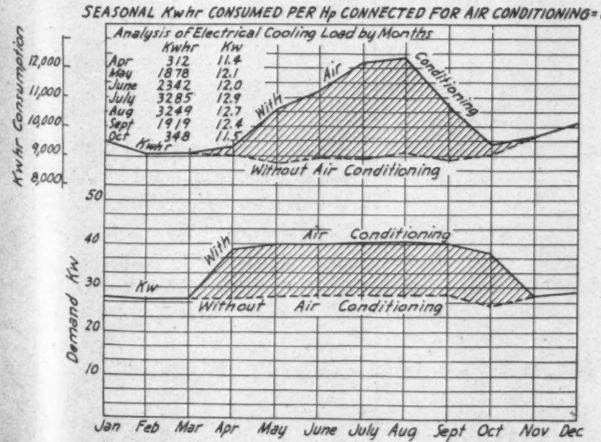
HOTELS

SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 785



SHOE STORES

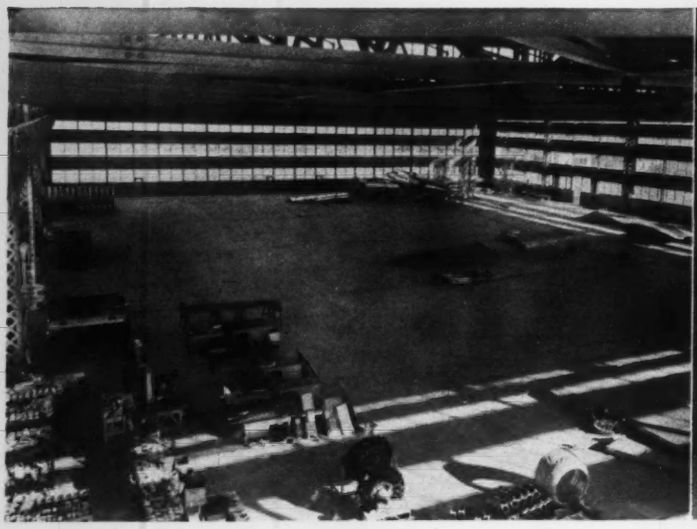
SEASONAL Kwhr CONSUMED PER Hp CONNECTED FOR AIR CONDITIONING = 675

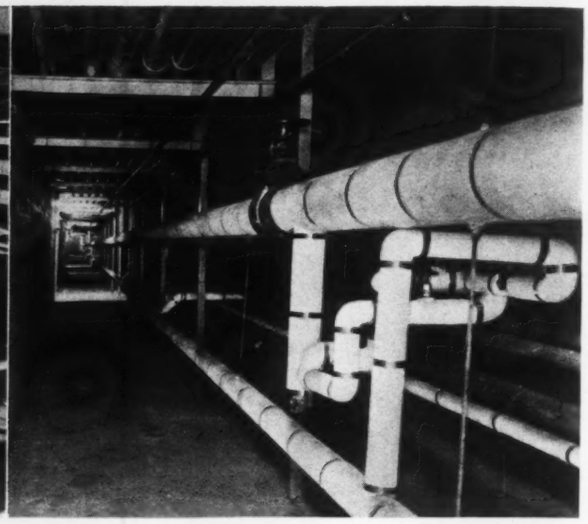


Electric power and light usage by months of the AVERAGE establishment where summer air conditioning has been installed, based on meter readings. Shaded areas show air conditioning "responsibility."

See "Electricity Used by Air Conditioning," by William P. Rock

HEATING, PIPING AND AIR CONDITIONING, April, 1937





***Left*—The new Boeing Aircraft assembly plant, which is 204 by 304 ft with a clear height of 35 ft. *Right*—All electrical lines, water lines, steam lines, sewer lines and sprinkler mains are in a 7x7 ft tunnel running the full length of the building just outside the building line. Smaller lateral trenches lead into the building, and hinged cast iron floor plates permit servicing of the lines**

Accessibility of Service Piping Features Aircraft Assembly Plant

ACCESSIBILITY of all service facilities is a feature of the new plant recently completed at Seattle, Wash., for the Boeing Aircraft Co. for assembly of its new 4-engine YB-17 Army bombers. The assembly plant is 204 ft by 304 ft with a clear span of 35 ft over the area which is entirely without obstruction.

Electricity, steam, compressed air, water and gas lines serving this huge open area of 2,100,000 cu ft enter the building through a large underground concrete tunnel leading from a nearby power house. Six tributary tunnels branch from this main tunnel beneath the floor with access openings every 20 ft and outlets for electricity and compressed air connections every 40 ft. The main tunnel is 360 ft long, 7 ft wide, and 6 ft 9 in. high, situated along side the building just outside the building line, so that its concrete roof would be flush with the adjoining floor if the building should be expanded in that direction. Inside the tunnel are a 5 in. steam main, a 3 in. air main, a 4 in. water main, a 1½ in. gas main, a sewer line, telephone lines, and conduit for wiring. The smaller concrete tunnels, 15 in. wide and 2 ft high, carry compressed air, electricity, steam return lines from the heating system, and water piping.

Three phase, 208 volt circuits serve oil and gasoline pumps at the corners of the building. One 3 hp pump is used to pump fuel oil from tank cars to a 12,000 gal fuel storage tank buried beside the power house. In conjunction with the pumping apparatus, a steam hose connection is provided which may be attached to the heating coils of tank cars to raise the temperature of the oil sufficiently for ease of flowing. Another 3 hp pump is used in connection with an 8,000 gal gasoline tank buried alongside the apron and connected to a gasoline hose pit at the end of the apron.

The heating system of the new plant centers in a 250 hp boiler with a steam atomized oil burner. A steam driven pump brings the fuel oil from the storage tank. Compressed air equipment is also provided for starting the furnace—atomizing the oil with air in place of steam. A steam operated boiler feed pump, automatically controlled, maintains a constant water level in the boiler. Steam is carried through the tunnel system to 12 blower type unit heaters within the assembly building, operating at 100 lb steam pressure, and each calculated to be equivalent to 2,000 sq ft radiation. Equipped with a ¾ hp motor driving three fans, each unit heater delivers 4500 cfm of air. There are three of the heaters on each side of the building, two at the rear wall, two over the front doors, and the remaining two mounted on roof trusses, all properly directed to furnish a complete circulation of heat over the large floor area. Despite the fact that the building contains 2,100,000 cu ft of unobstructed working space, it can be brought to working temperature by means of this heating system in an hour's time in moderate weather. Operating at full capacity, the heating system for the present building requires only 173 of the boiler's 250 hp.

Fire protection in the building is provided by a complete automatic sprinkler system including a 150,000 gal emergency sprinkler tank which may be used in the event of failure of the city water supply. A water pump, driven by a 125 hp motor, automatically switches on and pumps water from the emergency tank into the sprinkler system when the pressure in the sprinkler main drops to a predetermined low point.

The Austin Co. was the designer and builder.

The Strength and Design of Covers and Flanges for Pressure Vessels and Piping

[Part 6]

By T. McLean Jasper*, H. Gregersen*, and A. M. Zoellner*

THE construction of a vessel or pipe line for high pressure service involves a balance in the design at every point so that the safety of such equipment can be assured. Previous investigations have established the necessity for and the required amount of reinforcement for openings in vessel walls and for the proper shapes of heads of vessels. The present investigation aims to go into the question of covers and flanges of openings. Part 1, published in November, was concerned with covers and ring flanges; Parts 2 and 3, December and January, with hubbed flanges; Part 4, February, with bolt loads and Appendix 1 on the test arrangement; and Part 5, March, with Appendix 2 on test results. See page 176, March issue, for list of symbols used.

APPENDIX 3—FILLET COEFFICIENTS

As mentioned in the report the influence of the fillet on the strength of hubbed flanges is not provided for in the formulas by Holmberg-Axelson and Timoshenko. Generally, the problem of determining the influence of a fillet on the strength is extremely difficult, and in most cases, the determination of the proper size and shape of a fillet is based on the designer's experience or on approximate methods. Since the influence of the fillet on the strength of hubbed flanges has been found to be of rather major importance, it is desirable to have some method of determining this influence numerically. In the following, approximate fillet coefficients have been calculated to be used in connection with the formulas for hubbed flanges by Holmberg-Axelson and Timoshenko.

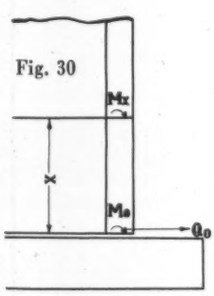


Fig. 30

Theoretical investigations have established that the bending moment M_x in the hub decreases rapidly with increased distance x from the joint between ring and hub. According to Holmberg-Axelson:

$$M_x = e^{-\beta x}\left[\left(\frac{-Q_o}{\beta}+M_o\right)\sin\beta x + M_o\cos\beta x\right] \quad \text{.....[28]}$$

$$\text{where: } \beta = \sqrt[4]{\frac{12\,(1-\mu^2)}{t^2d^2}} \quad \text{.....[29]}$$

when the directions of M_o and Q_o are chosen as shown in Fig. 30. Making the assumption that the flange is very rigid in the plane perpendicular to the axis, the radial displacement of the joint between ring and hub can be assumed to be zero and according to Timoshenko:

$$Q_o = \beta M_o \quad \text{.....[30]}$$

From Equations 28 and 30 it follows:

$$M_x = (e^{-\beta x}\cos\beta x)\,M_o \quad \text{.....[31]}$$

The variation of the ratio M_x/M_o with the distance x according to Equation 31 is plotted in Figs. 31 and 32 for the two types of hubbed test flanges dealt with in Chapter 3 of the report. (December, 1936, issue.)

From Figs. 31 and 32 it can be seen that the effect of a large fillet is to remove the location of failure from the junction between ring and hub to a certain location where the magnitude of the bending moment causing fracture is considerably smaller. This has been verified by means of the test results obtained with hubbed flanges provided with fillets showing that the strength increases with the size of the fillet, and that the fracture takes place in the part of the fillet close to the hub.

Furthermore, it can be seen from Fig. 31 that the strength of a flange with thin hub will increase more rapidly with increased fillet, than the strength of a flange with thick hub. This has been verified by means of the test results shown in Figs. 13 and 14.

It should be mentioned that the curves in Figs. 31 and 32 showing the variation of the moment M_x with the distance x (See Fig. 30) are based on constant hub thickness. If the variation of the hub thickness due to the fillet had been taken into consideration, the curves would have had a different shape. Since, however, an exact solution leads to results impractical for design purposes it has been assumed that a certain fraction n of the fillet radius is effective. From a study of the test results it was decided to assume:

$$n = 0.6\,R \quad \text{.....[32]}$$

as a rather conservative value leading to safe results in the determination of the influence of the fillet radius on the strength of hubbed flanges.

Assuming Poisson's ratio $\mu = 0.25$, Formula 29 can be written:

$$\beta = \frac{1.83}{\sqrt{td}} \quad \text{.....[33]}$$

From Equations 31, 32 and 33:

$$M_x = \left[e^{-\left(1.1\frac{R}{\sqrt{td}}\right)}\cos\left(1.1\frac{R}{\sqrt{td}}\right)\right]M_o \quad \text{.....[34]}$$

and when denoting:

$$\emptyset = e^{-\left(1.1\frac{R}{\sqrt{td}}\right)}\cos\left(1.1\frac{R}{\sqrt{td}}\right) \quad \text{.....[35]}$$

it follows:

$$M_x = \emptyset M_o \quad \text{.....[36]}$$

where M_o is the bending moment calculated from Holmberg-Axelson formulas or Timoshenko's formula, M_x is the bending moment producing the maximum stress, and $\emptyset$ a fillet coefficient depending on the fillet radius, the hub diameter and the hub thickness.

In order to facilitate the practical application of the fillet coefficient $\emptyset$ the Equation 35 has been numerically evaluated within the practical range and corresponding values of $\frac{R}{\sqrt{td}}$ and $\emptyset$ are given on page 674, December issue, from which the curve shown in

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Illustrations reproduced by courtesy of the A. O. Smith Corp.

This paper was presented at the annual meeting of the *American Society of Mechanical Engineers*, December, 1934, New York, N. Y., but has not heretofore been available in published form.

Fig. 8 has been plotted for the purpose of eliminating interpolations.

As before mentioned, the hubbed test flanges broke in the part of the fillet close to the hub. Neglecting the small additional thickness due to the fillet we may write

$$M_s = \frac{St^2}{6} \quad \ldots\ldots [37]$$

From Formulas 36 and 37 it follows:

$$S = \emptyset \frac{6}{t^2} M_o \quad \ldots\ldots [38]$$

where M_o can be determined from Holmberg-Axelson's and Timoshenko's formula. Combining Formula 39 with Timoshenko's Formula 22 results in:

$$S = \emptyset \frac{1.91M}{td\left(t + \frac{0.915t}{\sqrt{td}}T + \frac{1.257 \log_{10} D/d}{t\sqrt{td}}T^3\right)} \quad \ldots\ldots [39]$$

which is a convenient and sufficiently accurate formula for determination of the strength of hubbed flanges provided with a circular fillet.

APPENDIX 4—LOCATION OF GASKET

In the investigation of the test flanges described in this report, the external moment producing the maximum stress in the joint between ring and hub equals the load W multiplied by the distance a between the two concentric load circles. Thus the external moment for these test flanges will be Wa regardless of the location of the two load circles. In an actual flange subjected to internal pressure, gasket load and bolt load, the conditions are different. As can be seen from Fig. 6 the external moment depends a great deal on where the gasket is located. Thus if the gasket is located close to the bolt circle the pressure acting on the area $\pi/4\ (g^2 - d^2)$ of the flange must be taken into consideration in the determination of the external moment producing the maximum stress. Only when the gasket is located close to the inner edge of the flange ($g \cong d_m$) we may say that the external moment equals Wa. In the following the variation of the external moment with the location of the gasket, and the most favorable location of the gasket, will be investigated.

The flange shown in Fig. 33 is subjected to the external forces P, G, Q and W. The force P is assumed to be acting in the middle of the hub and its magnitude is:

$$P = \frac{\pi}{4} d_m^2 p \quad \ldots\ldots [40]$$

since the distance x is small compared to the diameter d_m of the hub the force Q which is the resultant of the pressure acting on the surface $\pi/4(g^2 - d_m^2)$ of the flange can be assumed to be acting in the middle of the distance x. The magnitude of Q will be:

$$Q = \pi x\ (d_m + x)\ p \quad \ldots\ldots [41]$$

The magnitude of G depends on the width of the gasket and the gasket pressure required to obtain sealing. The equation of equilibrium is:

$$W = P + Q + G \quad \ldots\ldots [42]$$

The external moment M producing the maximum stress in the joint between ring and hub will be:

$$M = W\ (a - x) + Q\frac{x}{2} + Px \quad [43]$$

Introducing 42 in 43

[*Concluded on page* 246]

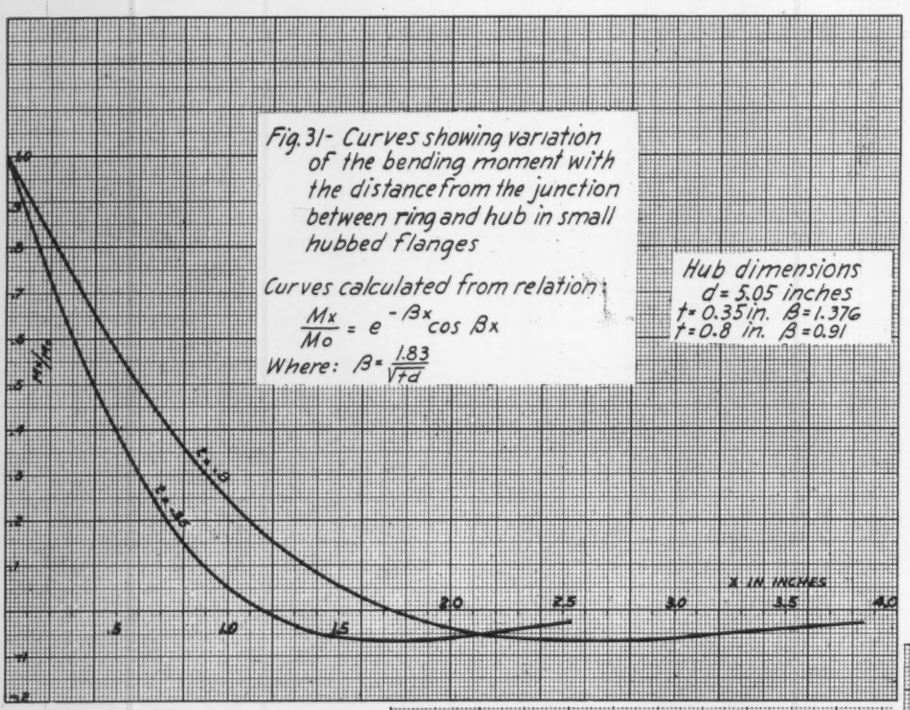


Fig. 31- Curves showing variation of the bending moment with the distance from the junction between ring and hub in small hubbed flanges
Curves calculated from relation:
Mx/Mo = e^-βx cos βx
Where: β = 1.83/√td
Hub dimensions
d = 5.05 inches
t = 0.35 in. β = 1.376
t = 0.8 in. β = 0.91
X IN INCHES

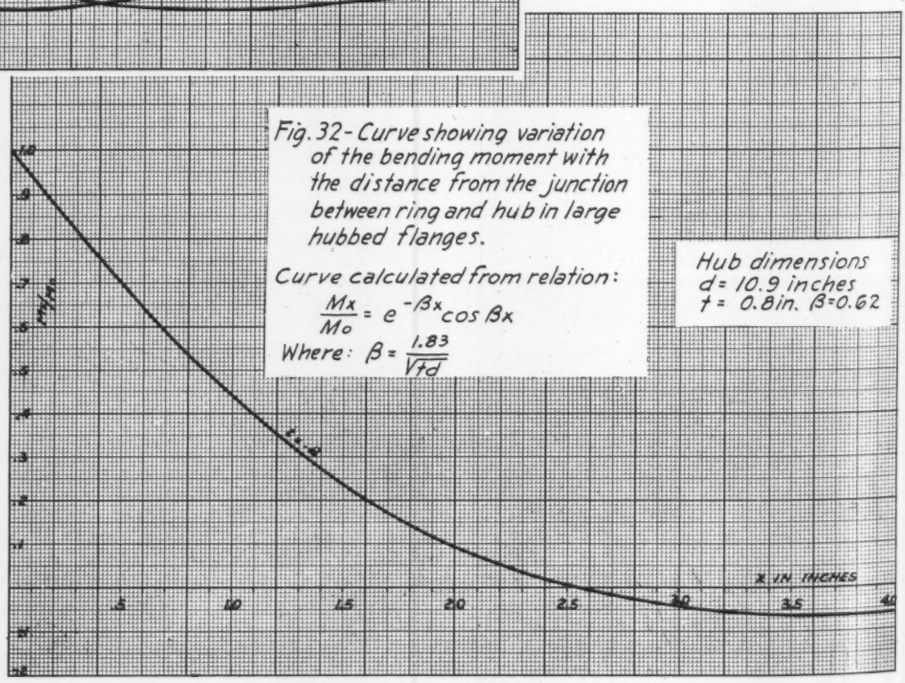


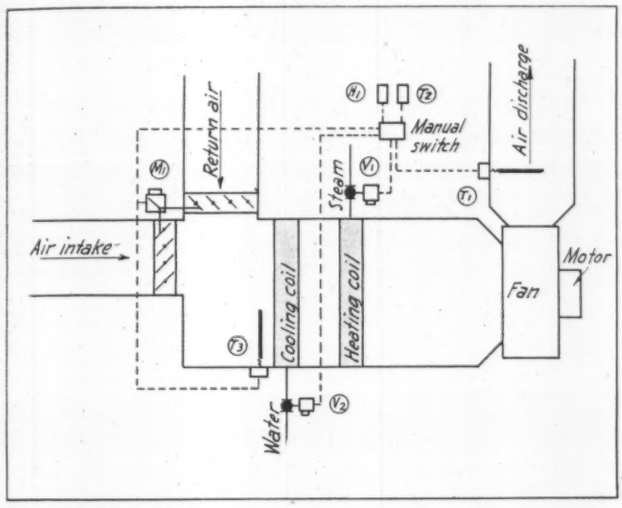
Fig. 32- Curve showing variation of the bending moment with the distance from the junction between ring and hub in large hubbed flanges.
Curve calculated from relation:
Mx/Mo = e^-βx cos βx
Where: β = 1.83/√td
Hub dimensions
d = 10.9 inches
t = 0.8 in. β = 0.62
X IN INCHES

CONTROLS

By C. L. Ringquist*
Reference Sheet 8

HEATING, VENTILATING, DEHUMIDIFYING AND COOLING SYSTEMS

One of the most simple control systems of this type:



Cycle—For summer operation, damper motor M_1 is set to introduce a predetermined amount of outside air. The manual summer-winter switch is in the summer position.

Room thermostat T_2 controls operation of modulating valve V_2 to allow more or less cooling as required. Room humidistat H_1 controls humidity by taking over control of motor or diaphragm operated valve V_2 whenever room relative humidity exceeds humidistat setting.

For winter operation, the summer-winter switch is in the winter position. Temperature controller T_3 controls operation of outside and recirculated air dampers through M_1 to maintain constant temperature of air entering heating coil.

T_2 now controls room temperature by modulating V_1 in steam supply line to heating coil.

Duct thermostat T_1 acts as a low limit control and takes control of V_1 to prevent air being introduced at too low a temperature.

The preceding example is a simple control method without accurate means of controlling humidity and temperature for the summer cycle. The following arrangement has incorporated into it an arrangement of cooling coils to control both the temperature and humidity and an arrangement of heating coils and dampers to give accurate control on the heating cycle:

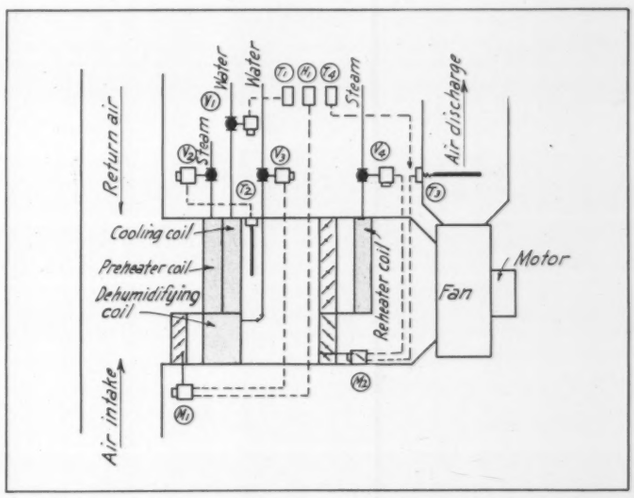
Cycle—The preheating and reheating coils are not in operation during summer operation. During winter operation the damper in front of the dehumidifying coil is closed because the humidity will always be lower than that maintained during the summer cycle.

In winter, the air mixture passes through the preheating coil where it is always heated to a constant leaving temperature. This heated air then passes either through the reheating coil where additional heat is added or part of the air passes through the bypass damper and is mixed with the reheated air before entering the fan.

For summer operation, room humidistat H_1 controls room humidity by controlling amount of air passing through dehumidifying coil by means of damper motor M_1 and face damper. M_1 is also equipped with an auxiliary switch to close off refrigerant supply to dehumidifying coil when no air is passing through coil; this is accomplished by means of the modulating motor or diaphragm valve V_3.

The temperature of the conditioned space is controlled by means of room thermostat T_1 which modulates supply of refrigerant to cooling coil through modulating motor or diaphragm valve V_1.

For winter operation, duct thermostat T_2 maintains constant temperature from the preheating coil by modulating steam supply to this coil through modulating valve V_2.



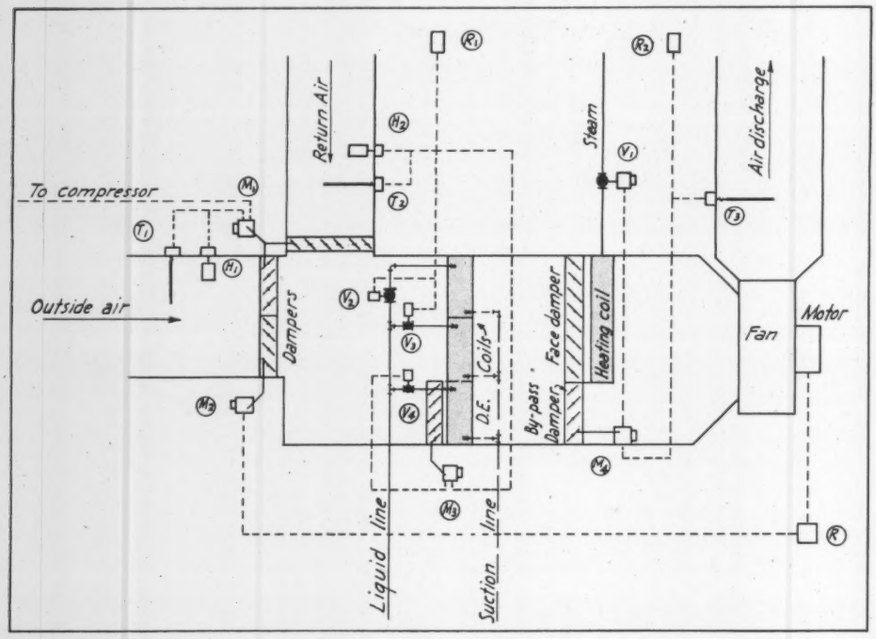
Room thermostat T_4 controls room temperature by regulating amount of air passing through reheating coil and through bypass. This is accomplished by means of modulating damper motor M_2, which is also under control of the low limit thermostat T_3 which assumes control if the leaving air temperature from the fan drops below a predetermined temperature. M_2 is also equipped with an auxiliary switch which closes the two position valve V_4 when all air is passing through the bypass, thus shutting off the steam supply to the reheating coil.

A summer-winter switch may be incorporated if desired to change the required controls from the cooling and dehumidifying cycle to the heating and ventilating cycle.

*Air Conditioning Department, The Trane Co., La Crosse, Wis.

Although data on automatic controls for heating, humidifying, dehumidifying, ventilating, cooling and complete air conditioning may be had from numerous sources, information on groups of controls applicable to various types of systems is not so readily available. The author has therefore diagrammed typical control layouts, with each of which is given an explanation of the control cycle. . . For the purpose of illustration, central fan coil types of systems are used; the control arrangements described are equally applicable to units built up to perform the same function. . . Types of systems covered by this series include humidifying; ventilating; heating and ventilating; heating, ventilating and humidifying; cooling, dehumidifying and ventilating; cooling, dehumidifying, ventilating and humidifying; heating, ventilating, dehumidifying and cooling; heating, ventilating, dehumidifying, cooling and humidifying; and zone control systems.

The following system incorporates additional refinements. For example, a set of controls and dampers is provided in the outside air and return air intake so that when the outside air is at such a condition that it may be used for cooling purposes, all outside air is introduced and the refrigeration compressor is shut down:



Cycle—The controls for winter operation include a room thermostat R_2 controlling amount of air passing through or around heating coil by means of damper motor M_4 and the face and bypass dampers, thus governing the room temperature. Duct thermostat T_3 acts as a low limit control to prevent introduction of low temperature air. Damper motor M_4 is equipped with an auxiliary switch to close two position motor or diaphragm valve V_1 whenever face damper in front of heating coil is entirely closed, thus preventing "drift" and unnecessary use of steam.

The controls for summer, or cooling and dehumidifying, operation include duct thermostat T_1 in the outside air intake and duct humidistat H_1 in the outside air intake, both of which must be satisfied before damper motor M_1 closes the recirculating damper and opens the "all outside air damper." M_1 is also equipped with an auxiliary switch which acts to shut off the refrigeration machines through a magnetic starter when the recirculating damper closes.

Duct humidistat H_2 controls humidity by allowing more or less air to pass through dehumidifying coil, this action being controlled through modulating damper motor M_3 and the face damper in front of the dehumidifying coil. Duct thermostat T_2 assumes control of the damper motor to prevent too low temperatures on mild days.

Damper motor M_3 is equipped with an additional contact to close refrigerant valve V_4 when the damper in front of the dehumidifying coil is entirely closed.

Room thermostat R_1 controls room temperature by opening or closing V_2 and V_3 in sequence whenever required.

Relay R in fan motor circuit closes outside air damper by means of M_2 when the fan stops, and opens outside air damper whenever fan starts. R acts both in summer and winter operation.

[*Concluded from page* 244]

$$M = Pa + Qa + Ga - Q\frac{x}{2} - Gx \quad \ldots\ldots\ldots\ldots\ldots\ldots [44]$$

Introducing 41 in 44

$$M = \pi p x\ (d_m + x)\ (a - \frac{x}{2}) - Gx + (P + G)a \ldots [45]$$

Equation 45 expresses the external moment in the most general way and should be applied in determination of the maximum stress by means of Timoshenko's formula.

The equation 45 will now be applied to two extreme cases:

Case 1—The gasket is located close to the inner edge of the flange. It will be assumed that $g = d_m$ (Fig. 33). In this case $x = 0$.

$$M = (P + G)\ a = Wa \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [46]$$

Thus, in this extreme case only, the external moment will be equal to Wa.

Case 2—The gasket is located close to the bolt circle. It will be assumed that $g = b$ (Fig. 33). In this case $x = a$ which introduced in Equation 45 results in:

$$M = \frac{\pi a^2}{2}\ (d_m + a)\ p + Pa \quad \ldots\ldots\ldots\ldots\ldots\ldots [47]$$

If Equation 47 results in a smaller value of the external moment M than Equation 46 it will be best to locate the gasket close to the bolt circle and if it results in a larger value it will be best to locate the gasket close to the inner edge.

The criterion for the best location of the gasket can be found in combining Equations 46 and 47 as follows:

$$(P + G)\ a \gtrless \frac{\pi a^2}{2}\ (d_m + a)\ p + Pa$$

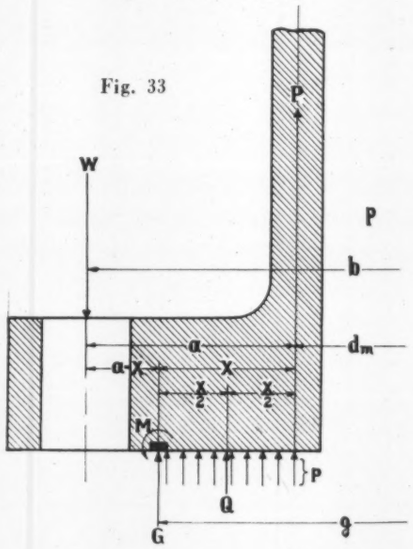
from which:

$$G \gtrless \frac{\pi a}{2}\ (d_m + a)\ p \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [48]$$

From Equation 48 it can be seen that the best location of the gasket depends mainly on the magnitude of the gasket load. If G is larger than the right side of Equation 48, the best location of the gasket will be close to the bolt circle, and if smaller, the best location of the gasket will be close to the inner edge of the flange.

[*To be concluded*]

Fig. 33



Tables of Pipe Properties Comprise This Month's Heating ✦ Piping and Air Conditioning Summary Sheets

Properties of Pipe†

By Arthur McCutchan*

The American Standard for Wrought Iron and Wrought Steel Pipe approved as a tentative standard by the *American Standards Association* in November, 1935, contains tables of dimensions and nominal weights of pipe covering a wide range of service conditions. The schedule numbers used approximate values of $1000 \times P/S$. Thus Schedule 40 is roughly suitable for a service pressure of 400 lb per sq in., provided an allowable stress of 10,000 lb per sq in. is satisfactory for the particular pipe material under the type of service and temperature conditions involved.

Since a single schedule of pipe wall thicknesses may cover the entire range of pipe sizes in a given service, the dimensional properties of pipe used in computing steam and water velocities, pressure drops and flexural characteristics of the lines can be conveniently grouped in tables for each schedule number.

An eight-bank calculating machine was used in computing the values for all but the pipe thicknesses which correspond to "standard weight" and "extra strong" designations. The properties of pipe for these latter wall thicknesses were taken from existing tables. The intermediate figures used in determining the moment of inertia, inside diameter to the fifth power, and the like, were carried out to eight places to insure numerical accuracy, although commercial variations in wall thickness render extreme accuracy meaningless. The properties of wrought iron pipe differ somewhat from the computed values for steel pipe given but the differences are well within the permitted manufacturing tolerance. Under-scored wall thicknesses are identical with thicknesses for "standard weight" pipe in former tables.

SIGNIFICANCE OF PROPERTIES TABULATED

Inside Diameter—The inside diameter, sometimes referred to as the "actual inside diameter," is used directly in solving pressure loss problems by the aid of such charts as that based on Fritzsche's formula for the flow of superheated steam. It also is used in determining the friction factor f in the "rational" or general flow formulas which are based on functions of the Reynolds number. Care must be exercised in determining this friction factor to insure that the inside diameter is properly expressed in feet or inches as the particular formula requires. Under turbulent flow conditions it is useful to note that the pressure drop for a given velocity of flow varies inversely as the inside diameter.

Inside Diameter Squared—The inside diameter squared appears in Clavarino's formula for stress in thick walled pipes. In formulas for viscous flow the pressure drop for a given velocity varies inversely as the square of the inside diameter of the pipe.

Inside Diameter to Fifth Power—The inside diameter to the fifth power appears in the "rational" or general flow formulas where volume or weight of flow is used and in the majority of the empirical formulas for flow of steam, water, oil and gas. Fritzsche's formula for superheated steam flow employs the inside diameter to the 4.97 power, while Weymouth's formula for flow of gas contains the inside diameter to the 5.33 power. With these minor exceptions, the pressure loss in pipe where flow is expressed in volume or weight of fluid can be considered to vary inversely as the inside diameter to the fifth power.

Section Modulus—Where needed for computing the strength of pipe used for structural supports, the section modulus S can be obtained from the relation $S = 2\ I/D$ since values for D and I are given.

Area of Metal—The cross-sectional area of metal enters into the calculation of the effects of end thrust imposed by constraint of thermal elongation of the line.

*Engineer, Engineering Division, The Detroit Edison Co., Detroit, Mich.

†Summarized from two articles published in HEATING, PIPING AND AIR CONDITIONING, October and November, 1936. For full information, refer to these articles.

Properties of Schedules 10, 20 and 30 Pipe

Nominal Pipe Size In.	Outside Diam In.	Wall Thickness In.	Inside Diam In.	Inside Diam Squared d^2	Inside Diam Fifth Power d^5	Area of Metal	Internal Cross-Sectional Area	External Surface	Moment of Inertia	Weight per Ft of Pipe	Weight of Water per Ft of Pipe
						SCHEDULE - 10					
		t	d			Sq In.	Sq In.	Sq Ft	In.⁴	Lb	Lb
14 OD	14.0	0.250	13.50	182.25	448,403	10.80	143.14	0.994	255.3	36.8	62.03
16 OD	16.0	0.250	15.50	240.25	894,660	12.37	188.69	1.310	382.7	42.1	81.74
18 OD	18.0	0.250	17.50	306.25	1,641,307	13.94	240.53	1.670	542.1	47.4	104.21
20 OD	20.0	0.250	19.50	360.25	2,819,505	15.51	298.65	2.766	766.4	52.8	172.72
24 OD	24.0	0.250	23.50	552.25	7,167,030	34.36	533.74	3.706	1,315.0	63.5	231.85
30 OD	30.0	0.312	29.376	862.95	21,875,768	29.10	777.76	5.401	3,206.0	99.0	337.02
						SCHEDULE - 20					
8	8.625	0.250	8.125	66.02	35,409	6.79	51.85	0.3601	57.72	22.4	22.47
10	10.75	0.250	10.25	105.06	115,141	8.24	82.52	0.5731	113.7	28.1	35.76
12	12.75	0.250	12.25	150.06	275,855	9.82	117.86	0.8186	191.8	33.4	51.07
14 OD	14.0	0.312	13.376	178.92	426,185	13.42	140.52	0.975	314.4	45.7	60.89
16 OD	16.0	0.312	15.376	236.42	859,442	15.39	185.69	1.289	473.2	52.3	80.50
18 OD	18.0	0.312	17.376	301.92	1,583,978	17.34	237.13	1.647	678.2	59.0	102.77
20 OD	20.0	0.375	19.25	370.56	2,643,344	23.12	290.04	2.014	1113	78.6	183.67
24 OD	24.0	0.375	23.25	540.56	6,793,832	27.83	424.56	2.948	1942	94.7	185.95
30 OD	30.0	0.500	29.00	841.0	20,511,149	46.34	660.52	4.587	5,042	158	286.23
						SCHEDULE - 30					
8	8.625	0.277	8.071	65.14	34,248	7.26	51.16	0.3553	63.35	24.7	22.17
10	10.75	0.307	10.136	102.74	106,987	10.07	80.69	0.5603	137.4	34.3	34.96
12	12.75	0.330	12.09	146.17	258,304	12.87	114.80	0.7972	248.4	43.8	49.74
14 OD	14.0	0.375	13.25	175.56	408,394	16.05	137.88	0.9575	372.8	54.6	59.75
16 OD	16.0	0.375	15.25	232.56	824,801	18.41	182.65	1.268	562.1	62.6	79.12
18 OD	18.0	0.437	17.126	293.30	1,473,261	24.11	230.36	1.600	830.3	82.0	99.84
20 OD	20.0	0.500	19.0	361.00	2,476,099	30.63	285.53	1.969	1,457	105	122.87
24 OD	24.0	0.562	22.876	523.21	6,264,703	41.39	411.00	2.854	2,843	141	176.09
30 OD	30.0	0.625	28.75	826.56	19,642,160	57.68	649.18	4.508	6,224	197	281.30

Here is the second installment of the new series of "summary sheets," on this and the preceding page. The purpose of these sheets is to make available for quick reference for everyday use information needed in heating, piping and air conditioning work. . . . For complete information, refer to the original article from which these data were obtained, as the purpose of these summaries is not to take the place of the article, but rather to put the essentials into convenient reference form. . . . The Editors will appreciate your suggesting what type of material you'd like presented in this way. . . . Address HEATING, PIPING AND AIR CONDITIONING, 6 North Michigan Avenue, Chicago, Ill.

Internal Cross-Sectional Area—The internal cross-sectional area in square inches and square feet facilitates calculations of velocities of flow. The area in square feet gives directly the contents in cubic feet per foot of length.

External Surface—The square feet of external surface per foot of length is useful in calculating the total heat loss from the bare pipe surface when the loss per square foot is known or assumed for a given operating condition.

Moment of Inertia—The moment of inertia appears in all computations of bending stress in pipe. It is used in determining reacting forces set up by constraint of thermal elongation, and in estimating the rigidity of all piping structures.

Weight of Pipe per Foot—These pipe weights are taken from *A.S.A.* B36-10. Pipe weights are used in checking the cost of pipe, in computing shipping costs, and in providing means for handling the pipe from the railroad car to position.

Weight of Water per Foot of Pipe—In computing the dimensions of hangers and supports, it is customary to assume the pipe to be full of water, even if the lines are intended for superheated steam. The weight of water in the larger pipe sizes exceeds the weight of pipe.

Contents—Contents in cu ft per foot of length is given by the column "Internal Cross-Sectional Area, Sq Ft."

More of these tables will appear next month.

Properties of Schedule 40 Pipe

Nominal Pipe Size in.	Outside Diam in. D	Wall Thickness in. t	Inside Diam in. d	Inside Diam Squared d^2	Inside Diam Fifth Power d^5	Area of Metal Sq In.	Internal Cross-Sectional Area Sq In.	Internal Cross-Sectional Area Sq Ft	External Surface Per ft Sq Ft	Moment of Inertia in.^{4}	Weight of Pipe Per ft Lb	Weight of Water Per ft of Pipe Lb
1/8	0.405	0.068	0.269	0.0724	0.00141	0.072	0.0569	0.00040	0.106	0.001064	0.25	0.0250
1/4	0.540	0.088	0.364	0.1325	0.00639	0.125	0.1041	0.00072	0.141	0.003312	0.43	0.0449
3/8	0.675	0.091	0.493	0.2430	0.02912	0.167	0.1909	0.00133	0.177	0.007291	0.57	0.0830
1/2	0.840	0.109	0.622	0.3869	0.09310	0.250	0.3039	0.00211	0.220	0.01709	1.86	0.1317
3/4	1.050	0.113	0.824	0.679	0.3799	0.333	0.5333	0.00371	0.275	0.03704	1.14	0.2315
1	1.315	0.133	1.049	1.100	1.270	0.494	0.8639	0.00600	0.344	0.08734	1.68	0.3744
1 1/4	1.660	0.140	1.380	1.904	5.005	0.669	1.495	0.01040	0.435	0.1947	2.28	0.6490
1 1/2	1.900	0.145	1.610	2.592	10.82	0.799	2.036	0.01414	0.497	0.3099	2.72	0.8823
2	2.375	0.154	2.067	4.272	37.72	1.075	3.356	0.02330	0.622	0.666	3.66	1.454
2 1/2	2.875	0.203	2.469	6.096	91.75	1.704	4.788	0.03322	0.753	1.530	5.80	2.073
3	3.5	0.216	3.068	9.413	271.8	2.228	7.393	0.05130	0.916	3.017	7.58	3.201
3 1/2	4.0	0.226	3.548	12.59	562.2	2.680	9.888	0.06870	1.047	4.788	9.11	4.287
4	4.5	0.237	4.026	16.21	1,058	3.173	12.73	0.08840	1.178	7.233	10.8	5.516
5	5.563	0.258	5.047	25.47	3,275	4.304	20.01	0.1390	1.456	15.16	14.7	8.674
6	6.625	0.280	6.065	36.78	8,206	5.584	28.89	0.2006	1.734	28.14	19.0	12.52
8	8.625	0.322	7.981	63.70	32,380	8.396	50.03	0.3474	2.258	72.49	28.6	21.68
10	10.75	0.365	10.02	100.4	101,000	11.90	78.85	0.5475	2.814	160.7	40.5	34.16
12	12.75	0.406	11.938	142.5	242,470	15.77	111.93	0.7773	3.338	300.3	53.6	48.50
14 OD	14.0	0.437	13.126	172.3	389,638	18.61	135.32	0.9397	3.665	429.1	63.3	58.64
16 OD	16.0	0.500	15.000	225.0	759,375	24.35	176.72	1.2272	4.189	731.9	82.8	76.58
18 OD	18.0	0.562	16.876	284.8	1,368,820	30.79	223.68	1.5533	4.712	1172	105	96.93
20 OD	20.0	0.593	18.814	354.0	2,357,244	36.15	278.00	1.9305	5.236	1703	123	120.46
24 OD	24.0	0.687	22.626	511.9	5,929,784	50.31	402.07	2.7921	6.283	3424	171	174.23

Heating · Piping
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AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

JOURNAL SECTION

CONTENTS • APRIL, 1937

The Cooling and Heating Rates of a Room With Different Types of Radiators and Convectors, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick..... 251

Summer and Winter Air Conditioning by Low Pressure Steam, by G. F. Zellhoefer..... 265

The Inspiration of Society Membership..... 271

A.S.H.V.E. Research—A Page of News Notes and Comment, by Brewster S. Beach..... 273

News of Local Chapters..... 275

Candidates for Membership..... 278

Candidates Elected 280

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The Cooling and Heating Rates of a Room With Different Types of Steam Radiators and Convectors

By A. P. Kratz,* M. K. Fahnestock,** and E. L. Broderick† (*MEMBERS*)
Urbana, Ill.

This paper is the result of research sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *in cooperation with the University of Illinois.*

THE objects of the investigation were: (1) to determine the influence of different types of direct steam radiators and convectors on the rate of cooling of a room under winter service conditions after the steam supply to the heating units was turned off; (2) to determine the influence of different types of direct steam radiators and convectors on the rate of heating the same room from a stated minimum temperature after the steam supply to the heating units was turned on; (3) to correlate the results with the physical characteristics of the heating units.

The investigation was made with six units, consisting of two direct cast-iron radiators, two convectors with cast-iron heating units, and two convectors with nonferrous heating units. Although it should be recognized that the results of tests of this type, made under unsteady conditions, are influenced by the thermal capacity of the room and the contents of the room in which the tests are made, it is probable that the relative performances of the different units are independent of the test room, and that comparisons based on these relative performances are therefore valid. The actual magnitudes represented by the test results apply only to the room in which the tests were made and, with the exception of those obtained under equilibrium conditions, would require modification before being applied to other rooms and under different conditions.

Description of Apparatus

The room heating testing plant, in which the tests were conducted, has been described in detail in previous publications[1, 2, 3, 4]. A plan and an elevation section of the plant are shown in Figs. 1 and 2. In brief, the plant consists of a large cold room with the walls, floor and ceiling insulated with two layers of 3-in. corkboard; two test rooms 9 ft x 11 ft with 9 ft ceilings; refrigeration equipment for maintaining sub-zero temperatures in the cold room; a complete thermocouple system for observing temperatures; steam, hot-water and electric service for testing units with different heating mediums; an eupatheoscope for determining the equivalent temperature; and controllers and recorders for observing data over periods of continuous operation.

Since the test rooms are located in the corners of the cold room, as shown in Fig. 1, two walls of each test room are formed by the insulated walls of the cold room. The other two walls of each test room are exposed to the refrigerated space, thus simulating a corner room of a building with two exposed walls. These walls are of standard frame construction, consisting of 5/8-in. redwood siding, building paper, 3/4-in. thick tongue and groove yellow pine sheathing, 2 x 4 yellow pine studding, and 3/8-in. wood lath with 1/2-in. gypsum plaster. As shown in Fig. 2, each room is provided with an attic and basement space equipped with electric heaters for the purpose of regulating the air temperatures above the ceilings and below the floors. The ceilings are made of 3/8-in. wood lath and 3/8-in. gypsum plaster with no flooring in the attic. The floors are of standard 2 1/4-in. x 13/16-in. yellow pine flooring over building paper placed on 3/4-in. thick tongue and groove sub-floors. Each test room has one double window 4 ft-6 in. by 5 ft overall and one door 3 ft by 7 ft overall placed in the exposed walls.

The location of the direct expansion refrigerating coils in the cold room, and the ammonia compressor and condenser on the lower floor of the Mechanical Engineering Laboratory are also shown in Figs. 1 and 2. The refrigerating apparatus has sufficient capacity to maintain sub-zero temperatures in the cold room when the temperatures in the test rooms are approximately 70 F. Oscillating fans located beneath the refrigerating coils are used to direct cold air against the north wall of each test room at a velocity approximately equivalent to that of a 10 mph wind.

As all of the tests reported in this paper were conducted in the west test room of the room heating testing plant, hereafter, all references to a test room, unless

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[1] Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, S. Konzo, and M. K. Fahnestock, A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, p. 79.

[2] Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 475.

[3] Performance of Convector Heaters, by A. P. Kratz and M. K. Fahnestock, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 351.

[4] University of Illinois Engineering Experiment Station Bulletins Nos. 192 and 223, and Reprint No. 1.

For presentation at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Swampscott, Mass., June, 1937.

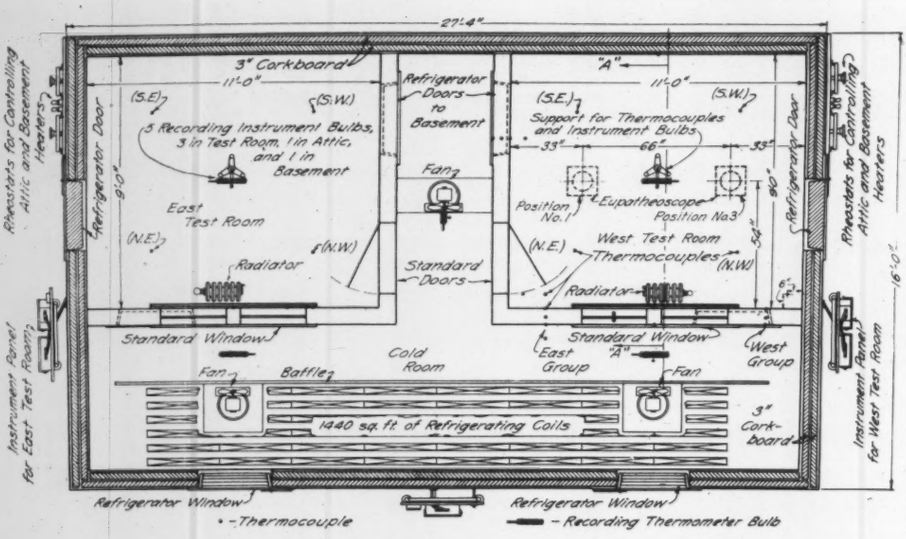

Fig. 1—Plan section of low temperature testing plant

otherwise stated, shall be understood to apply to the west test room. Fig. 2 shows the locations of some of the thermocouples available for observing air temperatures at various levels in the center of the test room and in the cold room, and surface temperatures on the walls, floor, and ceiling. The leads from thermocouples are formed into cables and connected to a common switchboard located on the lower floor of the laboratory directly beneath the cold room. The use of thermocouples permits the observer to note temperatures without entering the test room and thus disturbing the conditions, and they also make it possible to obtain surface and air temperatures in locations not accessible to thermometers.

The one-pipe steam connection for supplying steam to the units, and the condensate receiver, tank, and scales for determining the heat output of the units are shown in Fig. 2. Each unit was vented through a 1/8-in. pipe leading from the unit to the lower floor of the laboratory, and a constant rate of venting was maintained by means of an orifice 0.059 in. in diameter located in a pipe-cap on the discharge end of the vent pipe. Care was taken to make the vent connection to the units in such a manner that complete venting occurred without the loss of any unweighed condensate through the vent line.

The eupatheoscope, shown in Fig. 3, is an instrument used for evaluating the combined effect of radiation and convection in a heated space in which the air and the surfaces of the surroundings, such as radiators, furniture, walls, windows, etc., may be at different temperatures, in terms of an equivalent temperature consisting of the dry-bulb temperature of still air in a space in which the air and surroundings are at the same temperature. A complete description of this instrument, including a discussion of the rate of heat loss from the body and its correlation with human comfort, has been given in a previous publication.[5] Briefly, it consists of a sizeable body consisting of a blackened hollow copper cylinder mounted vertically on a stool, and inside of which is an electric heating element for maintaining the surface temperature of the cylinder. Thirteen thermocouples made of No. 34 B and S gage copper and constantan wire are embedded at regular intervals in the surface of the cylinder for observing the average temperature, which is maintained constant at 83.0 F by adjusting the electric power input to the heating element. From the power input, measured with a sensitive wattmeter, the heat loss from the copper cylinder may be determined in Btu per square foot per hour. By referring to the original calibration curve which was established by determining the unit heat loss from the cylinder when placed successively in a number of uniform environments having different dry-bulb temperatures, the combined effect of radiation and convection in the non-uniform environment can be expressed in terms of an equivalent temperature. This equivalent temperature consists of the dry-bulb temperature of a uniform environment which would give the same unit heat loss from the cylinder. For all of the tests, the eupatheoscope was located in position No. 3 as shown in Fig. 1.

The following six units used in the tests were regarded as being representative of characteristic types of com-

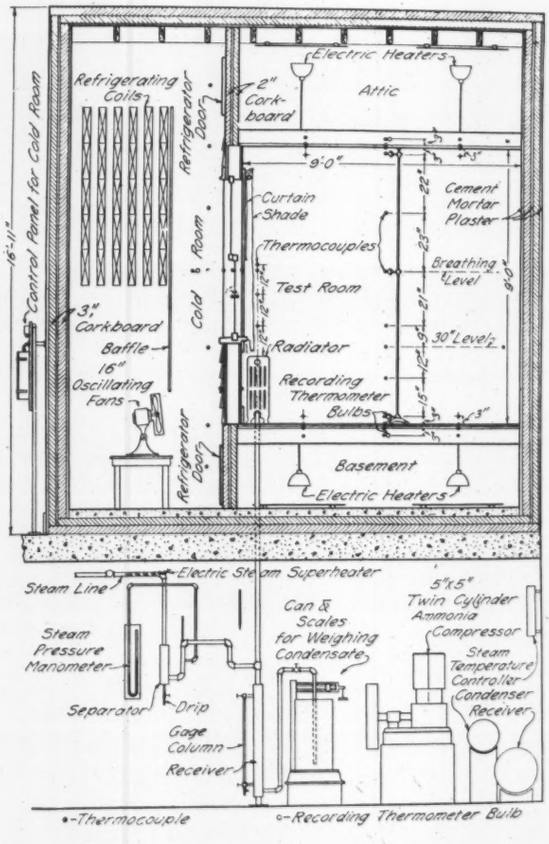

Fig. 2—Elevation section of low temperature testing plant

[5]The Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperature, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock, A.S.H.V.E. TRANSACTIONS, Vol. 39, 1933, p. 303.

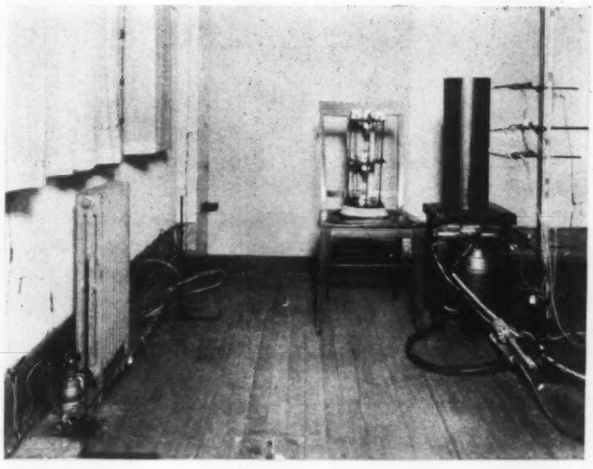

Fig. 3—The eupatheoscope with heating element removed

mercial units: Unit No. 1, an 8-section, 26-in., 5-tube, direct cast-iron radiator with 2½ in. between the centers of the sections; Unit No. 2, an 18-section, 25-in., 3-tube, direct cast-iron radiator with 1½ in. between the centers of the sections; Unit No. 3, a convector with a cast-iron heating element 6 in. high; Unit No. 4, a convector with a cast-iron heating element 14 3/16 in. high; Unit No. 5, a convector with a non-ferrous heating element 5¼ in. high; and Unit No. 6, a convector with a non-ferrous heating element 2 in. high. Additional information concerning the physical dimensions of each of these units is given in the insets in Fig. 4 and in Table 1.

In previous tests made in the room heating testing plant a temperature of 68 F at the 30-in. level in the test room was maintained under equilibrium conditions. However, for the purpose of these tests, in which the steam supply to the units was to be turned off until the temperature in the room dropped to 60 F, it was recognized that in order to bring this air temperature back to 68 F in a reasonable length of time, units with capacities slightly larger than those required for steady operation would be necessary. The units used were, therefore, selected with sufficient capacity to maintain as nearly as possible 71 F at the 30-in. level under equilibrium conditions.

Test Procedure

The test procedure with each unit was exactly the same and in general it may be divided into two parts; (1) that in which the conditions were in equilibrium or steady, and (2) that in which the conditions were unsteady due to the fact that the room was cooling or heating. In each case the unit was installed under the window in the exposed wall of the test room as shown in Fig. 2, and the temperatures in the cold room and attic were maintained constant at about minus 1.0 F and 62.0 F respectively. Whenever steam was supplied to the unit, the steam temperature was held constant at 216.5 F. The temperature maintained in the basement depended upon the air temperature 3 in. above the floor as established by each unit, and in each case it was regulated so that there was practically no heat loss or gain through the floor. Although no attempt was made to regulate these temperatures, external to the test room, during the portions of the tests over which the test room was heating up or cooling down, it was found that they did not vary an appreciable amount during these comparatively short periods of unsteady operation.

The surface temperature of the eupatheoscope cylinder was maintained at 83 F and the equivalent temperature of the environment in the test room was obtained for all operating conditions.

At the end of from 24 to 48 hours after a unit was installed in the test room, the air and wall temperatures were in complete equilibrium, and under these conditions of steady operation the following temperatures were observed and recorded: air temperatures in the cold room, attic and basement; air temperatures in the center of the test room at distances 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling; floor, ceiling and wall surface temperatures; and the temperature of the steam at the entrance to the unit. In the case of the radiators, the approximate temperature of the radiator itself was obtained by means of a thermocouple soldered to the surface of the center section, and in the case of the convectors, the approximate temperatures of the prime heating surface and the cabinet were obtained in the same manner. Simultaneously with these temperature observations, the net amount of steam condensed by the unit was determined at 10-min intervals over a period of one hour.

At the end of the equilibrium portion of the test the steam supply to the unit was turned off at the control valve on the lower floor of the laboratory, and successive sets of the temperatures previously enumerated were observed as rapidly as possible during the entire time that the room was cooling from the temperature existing at equilibrium. These observations were continued until the temperature at the 30-in. level reached 60 F. Care was taken to record the elapsed time from the beginning of the cooling period with each set of temperature observations.

When the temperature at the 30-in. level reached a minimum of 60 F the steam supply was turned on and again successive sets of temperature observations were made until the air temperature at the 30-in. level reached 68 F. The amounts of steam condensed in the unit over small intervals of time were obtained during the entire heating period for the purpose of determining the variation in condensing rate from the time that the steam supply was turned on until equilibrium conditions were practically reached.

A second cooling period, starting at the time that the steam supply was turned off when the temperature at the 30-in. level had reached 68 F, continued until the temperature had again dropped to 60 F. The temperature observations made during this cooling period were the same as those made during the first cooling period. The major difference in the two periods was in the degree of thermal equilibrium existing between the unit, the air, and the building materials in the walls, floor and ceiling of the room at the start of the period. In the first case the room and unit had been operating under steady conditions for a number of hours and the temperature at the 30-in. level was approximately 71.0 F, while in the second case the cooling period was preceded by two periods of unsteady operation, one with the room cooling, and one with the room heating, and the temperature at the 30-in. level was about 68 F. That is, the first

period was started under conditions of complete thermal equilibrium, while at the start of the second period such thermal equilibrium did not exist.

Following the second cooling period the steam supply to the unit was again turned on and for the remainder of the day and the following night the test plant, including the test room, was operated under steady conditions with the temperatures in the cold room, attic, and basement maintained constant at the values previously stated. The same routine of test, consisting of two cooling-down periods and one heating-up period, was then repeated for at least the second and sometimes the third day.

Results of Tests

In general the results may be discussed by considering the performance of the units under (1) equilibrium or steady conditions, and (2) under unsteady conditions.

The temperature gradients or the air temperatures in the test room 3 in., 30 in., and 60 in. above the floor

Table 1—Dimensions and Temperatures of Units, and Energy Available and Emitted During Equilibrium and Changing Conditions in First Cooling Period

Unit No.	Convector No.	Description of Unit	Overall Dimensions of Unit, Inches			Overall Dimensions of Heating Unit, Inches			Weight in Lb			Volume of Steam Space, Cu Ft	Weight of Saturated Steam in Unit in Lb at 216.5 F	Temperature in deg F Under Equilibrium Conditions with Steam Temperature at 216.5 F	
			Length	Height	Width	Length	Height	Width	Heating Unit	Cabinet	Total			Heating Unit	Cabinet
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1	.....	8-Sec., 26 in., 5-Tube Cast-Iron Radiator[a]	20	26	8	20	26	8	151.0	.....	151.0	0.3180	0.0129	214.0	.....
2	.....	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[b]	27	25	3½	27	25	3½	115.0	.....	115.0	0.2935	0.0119	214.5	.....
3	32	Convector with Cast-Iron Heating Unit...	34½	29	5⅞	33⅛	6	5¾	94.0	36.0	130.0	0.1448	0.0059	213.6	130.0
4	1	Convector with Cast-Iron Heating Unit...	37⅝	26	5⅝	36	14 3/16	5⅜	82.0	36.0	118.0	0.1463	0.0059	213.3	124.1
5	38	Convector with Non-ferrous Heating Unit.	36½	29	5⅜	35½	5¼	5	33.0	38.0	71.0	0.0508	0.0021	213.6	120.1
6	6	Convector with Non-ferrous Heating Unit.	30	28	5⅝	29⅝	2	5¼	13.0	30.0	43.0	0.0162	0.0007	212.7	112.3

Unit No.	Convector No.	Description of Unit	Heat Emitted to Room Under Equilibrium Conditions, Btu (Temp. of Steam = 216.5 F)				Heat Output of Units Based on Standard Equilibrium Conditions			Approximate Thermal Energy Available in Unit Above 60 Deg F When Steam Supply was Turned Off, Btu				
			From Heating Unit per Hour	From Eupathoscope per Hour	Total		Sq Ft of Equivalent Direct Radiation (E D R)	Weight of Metal, Lb per Sq Ft of Equivalent Direct Radiation		In Metal of Heating Unit	In Metal of Cabinet	Total in Metal of Heating Unit and Cabinet	In Steam in Heating Unit	Total in All Metal and Steam
					Per Hour	Per 10 Min		Based on Heating Unit	Based on Heating Unit and Cabinet					
1	2	3	17	18	19	20	21	22	23	24	25	26	27	28
1	.....	8-Sec., 26 in., 5-Tube Cast-Iron Radiator[a]	6308	68	6376	1063	28.1	5.37	5.37	2768	.....	2768	15	2783
2	.....	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[b]	6463	68	6531	1089	28.3	4.06	4.06	2114	.....	2114	13	2127
3	32	Convector with Cast-Iron Heating Unit...	6192	77	6269	1045	25.1	3.74	5.18	1718	285	2003	7	2010
4	1	Convector with Cast-Iron Heating Unit...	5833	78	5911	985	23.8	3.44	4.96	1496	261	1757	7	1764
5	38	Convector with Non-ferrous Heating Unit.	6173	77	6250	1042	25.0	1.32	2.84	537	258	795	2	797
6	6	Convector with Non-ferrous Heating Unit.	5959	88	6047	1008	23.8	0.55	1.81	210	177	387	1	388

Unit No.	Convector No.	Description of Unit	Temperatures in Deg F 10 Min After Steam Supply was Turned Off		Approximate Heat Emitted to Room During the 10 Min Period Following the Turning Off of the Steam Supply, Btu.						Temp. Diff. Between Floor and Ceiling 10 Min After Steam Supply was Turned Off
			Heating Unit	Cabinet	From Metal in Heating Unit	From Metal in Cabinet	Total from Metal in Heating Unit and Cabinet	From Steam in Unit	From Eupatheosope	Gross Total	
1	2	3	29	30	31	32	33	34	35	36	37
1	.....	8 Sec., 26 in., 5-Tube Cast-Iron Radiator[a]	167.7	.....	832	.....	832	13	13	858	19.5
2	.....	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[b]	157.7	.....	777	.....	777	12	13	802	18.7
3	32	Convector with Cast-Iron Heating Unit...	154.6	112.0	660	73	733	6	14	753	19.1
4	1	Convector with Cast-Iron Heating Unit...	147.0	108.5	647	63	710	6	15	731	17.4
5	38	Convector with Non-ferrous Heating Unit.	81.7	87.3	461	141	602	2	15	619	13.8
6	6	Convector with Non-ferrous Heating Unit.	66.8	81.3	201	105	306	1	18	325	15.3

[a]Sections on 2½-in. centers. [b]Sections on 1½-in. centers.

Table 2—Temperature of Air and Inside Surfaces of Exposed Walls Under Equilibrium Conditions

Unit No.	Description of Unit	Temperature of Air and Inside Surfaces of Exposed Walls Under Equilibrium Conditions, F										Difference in Temperature, F, Between Air at Center of Room and Inside Surfaces of Walls at the Same Level		Difference[a] Between Aver. of Temp. of Air at Center of Room and Aver. of Temp. of Inside Surfaces of Exposed Walls, F
		Air in Cold Room	Air at Center of Room				Inside Surface of Walls					30-in. Level	60-in. Level	3-in. Below Ceiling
			3-in. Above Floor	30-in. Level	60-in. Level	3-in. Below Ceiling	Difference between Floor and Ceiling, F	30-in. Level	60-in. Level	3-in. Below Ceiling	Back of Unit			
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
—1	8-Sec., 26 in., 5-Tube Cast-Iron Radiator[b]	—1.0	63.1	71.2	78.5	84.5	21.4	58.7	61.9	69.1	128.5	12.5	16.6	15.4	14.8
—2	18-Sec., 25 in., 3-Tube Cast-Iron Radiator[c]	—1.2	62.1	71.1	77.0	83.2	21.1	58.4	61.3	68.6	136.4	12.7	15.7	14.6	14.3
3	Convector with Cast-Iron Heating Unit	—1.1	60.9	70.8	79.1	84.8	23.9	58.1	61.7	68.5	82.6	12.7	17.4	16.3	15.5
4	Convector with Cast-Iron Heating Unit	—1.6	61.1	71.6	77.7	82.1	21.0	57.9	60.4	66.9	72.1	13.7	17.3	15.2	15.4
5	Convector with Non-ferrous Heating Unit	—1.3	61.0	71.1	79.4	84.3	23.3	58.2	61.7	68.4	76.0	12.9	17.7	15.9	15.5
6	Convector with Non-ferrous Heating Unit	—0.9	59.9	70.2	78.5	83.4	23.5	57.6	60.9	68.2	74.3	12.7	17.6	15.2	15.2

[a]Based on temperatures at 30-in. level, 60-in. level and 3-in. below ceiling.
[b]Sections on 2½-in. centers.
[c]Sections on 1½-in. centers.

and 3 in. below the ceiling are shown in Figs. 4a and 4c for the six different units tested under equilibrium conditions. The numerical values of the difference in temperature between the floor and ceiling obtained with the different units are given in col. 8, Table 2. Although the sizes or capacities of the units were limited to those commercially available, the temperatures at the 30-in. level with five of the units came within the range of from 70.8 to 71.6 F, a variation of less than one degree. The temperature with unit No. 6, which was a convector with a non-ferrous heating unit, was 70.2 F, a variation from the maximum 30-in. level temperature of 1.4 F. Thus, with the exception of the latter unit, the temperatures at the 30-in. level were practically the same.

For the purpose of comparison, the temperature gradients for unit No. 1, the cast-iron radiator with the sections on 2½-in. centers, are given in both Figs. 4a and 4c. The temperature gradient curves for units Nos. 3 and 4, the two convectors with cast-iron heating units, are shown in Fig. 4c. It may be observed that, with the same temperature at the 30-in. level, the temperature at the floor with the two convectors was about 2.0 F lower than it was with the radiator. Unit No. 4, as a result of a lower temperature at the ceiling, gave a slightly less overall difference in temperature between the floor and ceiling than that obtained with either unit No. 1 or unit No. 3. Fig. 4a, in addition to the temperature gradient curve for unit No. 1, includes the curves for units Nos. 2, 5, and 6. Unit No. 2 was a comparatively light weight radiator with sections on 1½-in. centers, and units Nos. 5 and 6 were convectors with non-ferrous heating units. As indicated by these curves and the data in col. 8, Table 2, the floor to ceiling temperature differences for the two radiators were approximately the same, but were slightly less than those for the two non-ferrous convectors, which were also practically alike.

Following the completion of each equilibrium test, the steam supply to the unit was turned off and the test room was allowed to cool until the temperature at the 30-in. level reached 60 F. Figs. 4b and 4d show the temperature gradients in the test room for each unit 10 min after the steam supply was turned off. These curves, obtained while the room was cooling, compared with the curves in Figs. 4a and 4c, obtained under equilibrium conditions, show that the temperature difference between the floor and the ceiling decreased when the supply of steam to the heating unit was turned off. A comparison of the numerical values of the temperature differences given in col. 8, Table 2, and col. 37, Table 1, shows that the average decrease in the temperature difference at the end of 10 min after the steam supply was turned off was 2.2 F for the radiators, 4.2 F for the convectors with the cast-iron heating units, and 8.9 F for the convectors with the non-ferrous heating units.

The chart and the accompanying data at the top of Fig. 4 show the net amount of steam condensed by each unit in pounds per hour under equilibrium conditions and the temperatures which existed during the tests. The small table gives the dry-bulb temperature at the 30-in. level, the equivalent temperature as determined by the eupatheoscope, and the difference between the two temperatures. The capacities of the units in square feet of equivalent direct radiation (EDR) based on the net amount of steam condensed corrected to standard con-

ditions with respect to steam and air temperatures and on a heat output of 240 Btu per square foot per hour are given in col. 21, Table 1. A comparison of the capacities of units Nos. 1 to 5 inclusive shows that when direct radiators were used the capacities required to heat the room to 71 F at the 30-in. level were slightly larger than they were when convectors were used. This apparent advantage of the convectors is offset by the fact that, although the dry-bulb temperatures produced by the units were the same, the equivalent temperatures obtained with the convectors were lower than those obtained with the direct radiators. The correlation between human comfort, dry-bulb temperature, and equivalent temperature has been discussed in a previous paper,[6] in which it was shown that if both radiators and convectors were of the proper size to maintain the same equivalent temperatures, the differences in the capacities based on the actual amounts of steam condensed were negligible.

Characteristic curves showing data obtained as relating to air temperatures and equivalent temperatures in the test room during the tests conducted with units Nos. 1, 4, and 5 are given in Figs. 5, 6, and 7. As these curves are representative of the performance of the three types of units tested, similar data obtained with the remaining radiators and convectors are not shown, except in Figs. 8 and 9 in which all performance curves are assembled in order to facilitate comparison. Referring to Figs. 5, 6, and 7, it may be noted that each figure has been divided into three parts, *a*, *b*, and *c*. In each case, part *a* pertains to the first cooling period when the steam supply to the unit was turned off at the completion of the equilibrium portion of the test and the room was allowed to cool until the temperature at the 30-in level reached 60 F; part *b* pertains to the heating period immediately following the first cooling period; and part *c* pertains to the second cooling period, which started at the end of the heating period at the time that the steam supply was turned off when the temperature at the 30-in. level reached 68 F. Curves showing the equivalent temperature in the test room are also included for all periods of the tests.

Referring to Figs. 5a, 6a, and 7a it may be observed that during the first portion of the cooling periods the temperatures of the air 3 in. below the ceiling decreased more rapidly than those at any of the other four levels in the room. In the case of the cast-iron radiator, unit No. 1, shown in Fig. 5a, the cooling near the ceiling continued at a rate greater than that at the other levels until the temperature at the 30-in. level reached approximately 63 F. This occurred about 45 min after the steam supply was turned off. From that time on until the end of the cooling period, the temperature of the air near the ceiling decreased more slowly than it did

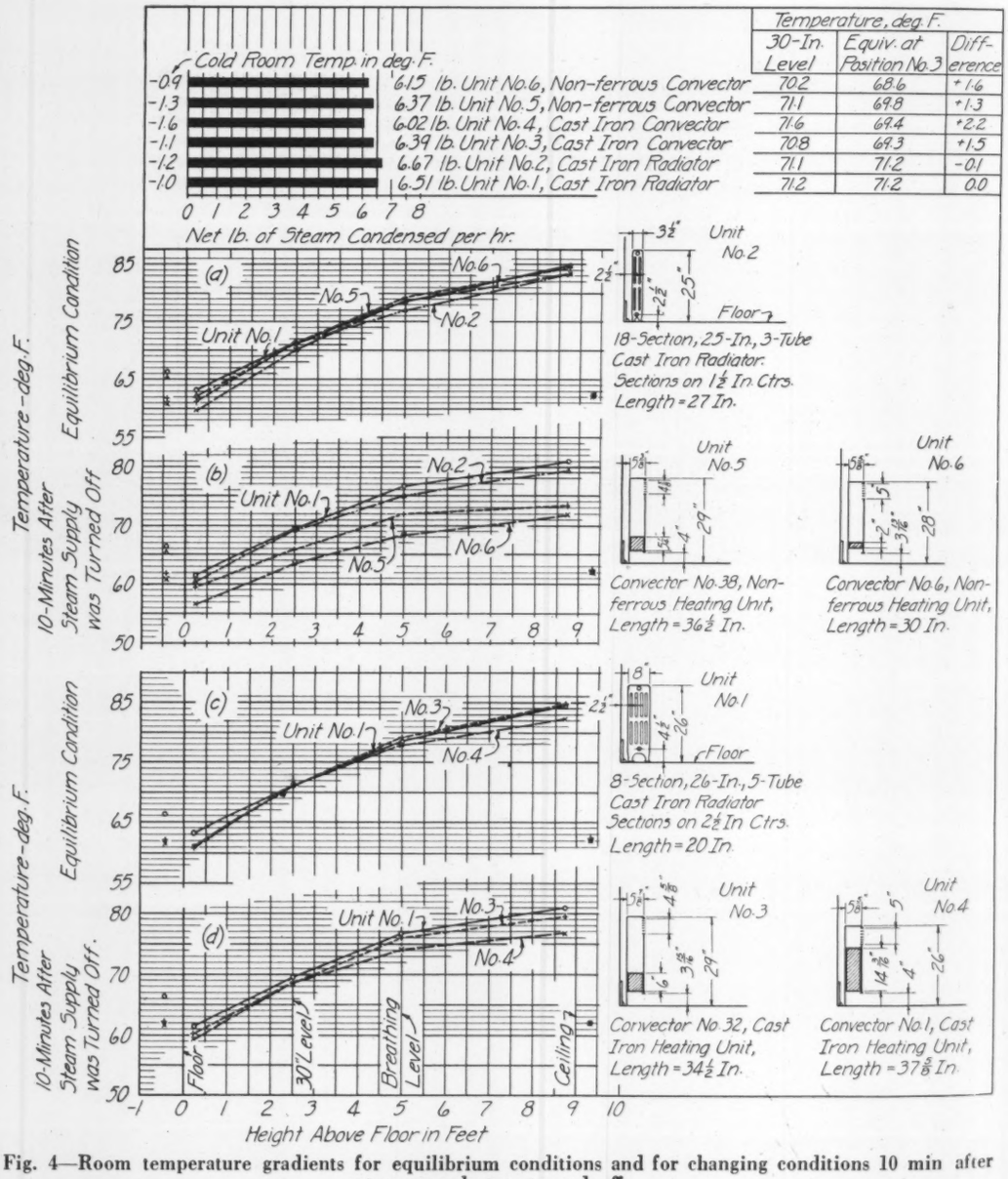
Temperature, deg. F.
30-In. Level
Equiv. at Position No. 3
Difference
70.2 68.6 +1.6
71.1 69.8 +1.3
71.6 69.4 +2.2
70.8 69.3 +1.5
71.1 71.2 -0.1
71.2 71.2 0.0
Cold Room Temp. in deg. F.
-0.9 -1.3 -1.6 -1.1 -1.2 -1.0
6.15 lb. Unit No. 6, Non-ferrous Convector
6.37 lb. Unit No. 5, Non-ferrous Convector
6.02 lb. Unit No. 4, Cast Iron Convector
6.39 lb. Unit No. 3, Cast Iron Convector
6.67 lb. Unit No. 2, Cast Iron Radiator
6.51 lb. Unit No. 1, Cast Iron Radiator
Net lb. of Steam Condensed per hr.
Temperature - deg. F. 10-Minutes After Steam Supply was Turned Off
Equilibrium Condition
(a) Unit No. 1 No. 2 No. 5 No. 6
(b) Unit No. 1 No. 2 No. 5 No. 6
(c) Unit No. 1 No. 3 No. 4
(d) Unit No. 1 No. 3 No. 4
Floor 30" Level Breathing Level Ceiling
Height Above Floor in Feet
Unit No. 2
18-Section, 25-In., 3-Tube Cast Iron Radiator. Sections on 1½ In. Ctrs. Length = 27 In.
Unit No. 5
Convector No. 38, Non-ferrous Heating Unit, Length = 36½ In.
Unit No. 6
Convector No. 6, Non-ferrous Heating Unit, Length = 30 In.
Unit No. 1
8-Section, 26-In., 5-Tube Cast Iron Radiator Sections on 2⅛ In. Ctrs. Length = 20 In.
Unit No. 3
Convector No. 32, Cast Iron Heating Unit, Length = 34½ In.
Unit No. 4
Convector No. 1, Cast Iron Heating Unit, Length = 37⅝ In.

Fig. 4—Room temperature gradients for equilibrium conditions and for changing conditions 10 min after steam supply was turned off

[6] Loc. Cit. See Note 5.

at the other levels in the room. Likewise, with units Nos. 4 and 5, Figs. 6a and 7a, the temperature of the air near the ceiling decreased more rapidly than it did at the other levels immediately following the start of the cooling period. As the cooling progressed, the rate at which the upper portion of the room cooled changed so materially that in both cases, during the latter part of the cooling period, the relationship between the rates of cooling in the upper and lower portions of the room was the reverse of what it was during the first part of the period. This change in the relative cooling rates of the two parts of the room, as shown by the temperature of the air 3 in. below the ceiling in Figs. 6a and 7a, occurred about 27 min after the steam supply was turned off from unit No. 4 and only 8 min after it was turned off from unit No. 5. An examination of the temperatures of the heating units at the time that the rate of cooling near the ceiling ceased to be greater than the rates at the other levels in the room, showed that in each case those temperatures were less than 100 F. This would seem to indicate that at this time the major portion of the thermal energy which was stored in the units when the steam supply was turned off had been dissipated into the room. No doubt the relatively high air temperatures occurring near the ceiling at the beginning of the cooling periods were brought about by heated air emitted directly from the units, and it is reasonable to assume that these temperatures would be the first to be affected when the steam supply was turned off. It is also reasonable to assume that the thermal energy stored in the units would have an appreciable effect on the room temperature gradients during the time that the stored energy was being transmitted to the room in quantities comparable to the heat loss from the room. As this supply of energy decreased, a point would be reached at which it no longer had an appreciable influence on the temperature gradient in the test room, and a readjustment of the relative rates of cooling in the room would probably occur. From then on the room would probably cool at a more uniform rate at all levels. This may serve as an explanation for the uniformity in the cooling rates that existed during the latter part of the cooling periods for the radiator and the two convectors, as shown in Figs. 5, 6, and 7.

The temperatures of the air at the four levels in the test room during the heating periods are given by the curves in Figs. 5b, 6b, and 7b. In each case, during the first part of the heating period, the temperatures near the ceiling increased more rapidly than those at the lower levels. Also, in every case, the rate of increase in the temperatures at all levels during the first part of

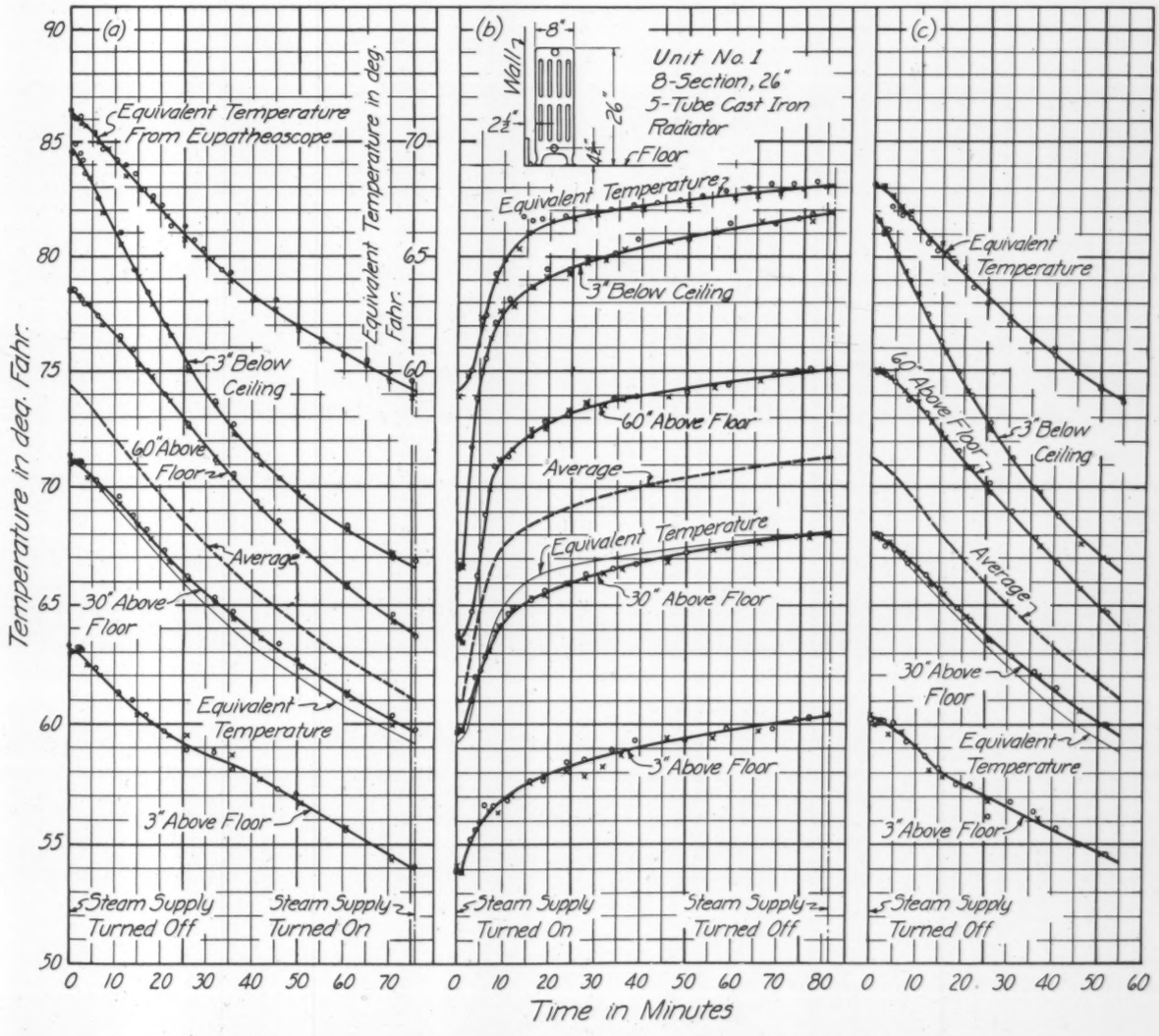

Fig. 5—Cooling and heating curves for room with Unit No. 1; 8-section, 26-in., 5-tube cast iron radiator

the heating period was equal to or greater than the rates of cooling at the corresponding level during the first part of the cooling period. These comparatively high heating rates continued for about 10 min after the steam supply was turned on, and at the end of that time there was a tendency for the temperature rise to be more gradual and uniform at all levels in the room.

differed from the first in that it was preceded by a period in which the conditions were changing. These tests were included because the changing conditions would more nearly simulate practical conditions present under intermittent operation with thermostatic control. In general, the result curves fall in the same order and appear to have the same relationship to each other as they did

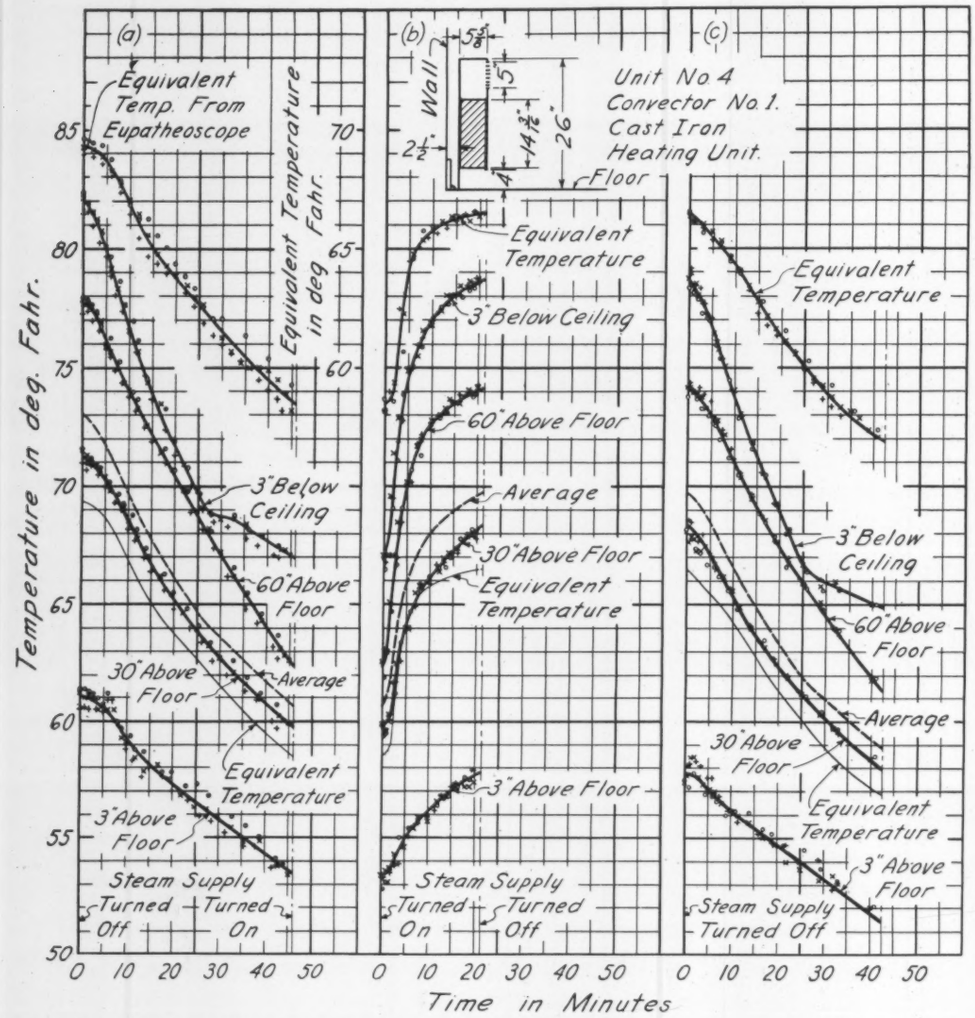

Fig. 6—Cooling and heating curves for room with Unit No. 4; convector with cast iron heating unit

Thus, during the first part of both the cooling and heating periods, with both the radiator and the convectors, the rate at which the temperature of the air decreased or increased varied at different levels in the room. The maximum rate of change occurred near the ceiling and this rate decreased progressively with the level in the room to a minimum occurring in the level near the floor. Also, as the cooling or heating progressed, the rates at which the temperature of the air was changing at different levels in the room became more nearly the same, so that during the latter part of each cooling or heating period, with both the radiator and the convectors, the air was changing in temperature at approximately the same rate at all levels in the room.

Curves giving the temperature of the air at the different levels in the room during the second cooling period are shown in Figs. 5c, 6c, and 7c. This cooling period during the first cooling period, and apparently the difference in the room and air conditions at the start of the two cooling periods did not materially affect the manner in which the room cooled. It is, therefore, assumed that the discussion devoted to the results presented for the first cooling period is also applicable to the results for the second cooling period.

The curves shown as broken lines in Figs. 5, 6, and 7 give the average air temperatures at the center of the test room. In each case they represent an average of the four full line curves which give the air temperatures 3 in., 30 in., and 60 in. above the floor and 3 in. below the ceiling. With all three units the average temperature at the start of the first cooling period fell between the temperatures at the 30-in. and the 60-in. levels. In the case of the radiator, Fig. 5a, it was more nearly midway between these two temperatures than it was in the case

of the two convectors, as shown in Figs. 6a and 7a. Referring to Figs. 4a and 4c, the relatively low floor and ceiling temperatures maintained with unit No. 4, and the relatively low floor temperature maintained with unit No. 5 serve to explain why the average temperature for these two convectors was relatively lower than it was for the radiator. Since the average temperature in the room with both the radiator and the convectors was always lower than that at the 60-in. level, it is evident that calculated heat losses from the room based on the temperature at the 60-in. level will tend to be too high, and the amount of the error will tend to be larger with the convectors than with the radiator.

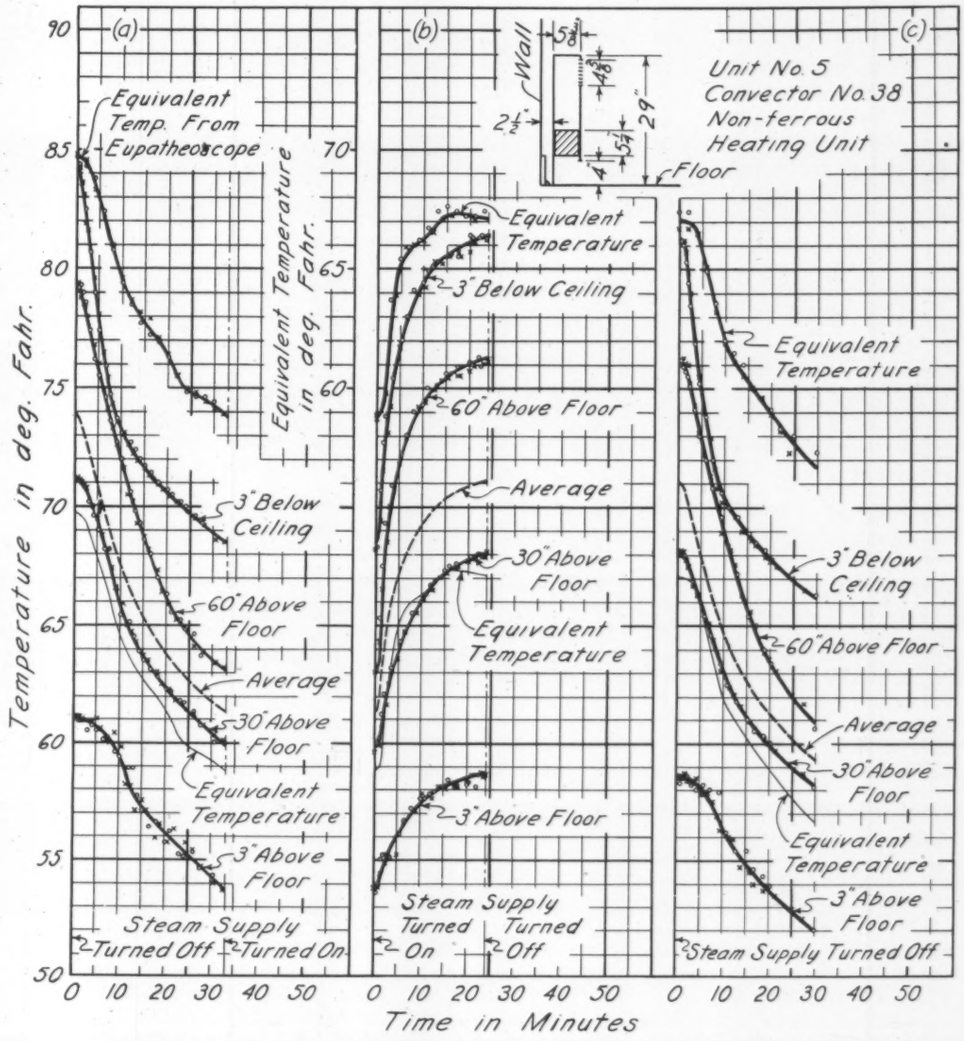

Fig. 7—Cooling and heating curves for room with Unit No. 5; convector with non-ferrous heating unit

In order to facilitate comparison between the equivalent temperature and the dry-bulb temperatures at the center of the room, the original equivalent temperature curves have been superimposed on the dry-bulb temperature scales and are shown as light-weight full lines in Figs. 5, 6, and 7. It may be observed that in all cases these equivalent temperature curves lie near the curves giving the temperature at the 30-in. level. At the start of the first cooling period, with the radiator, the equivalent and dry-bulb air temperatures at the 30-in. level were the same, while with both convectors the equivalent temperatures were about 1.5 F lower than the dry-bulb temperatures. This difference was previously mentioned under the discussion of the temperature gradient curves shown in Figs. 4a and 4c. As the cooling progressed in the case of the radiator, the equivalent temperature decreased at a relatively greater rate than that for the 30-in. level temperature so that during the latter part of the cooling period it was slightly lower than the latter temperature. In the case of the convectors, during the cooling period the difference between the equivalent and the 30-in. level temperatures decreased a small amount over that which existed at the beginning of the period. During the first part of the heating period, with both the radiator and the convectors, the equivalent temperature rose rapidy to a value above or at least equal to the 30-in. level temperature. However, as the heating period continued, the equivalent temperatures with the convectors became less than the 30-in. level temperatures and, with the radiator, the two temperatures became the same.

In Figs. 8 and 9 curves showing the temperatures obtained with all six of the units tested have been assembled. The temperatures 3 in. and 30 in. above the floor

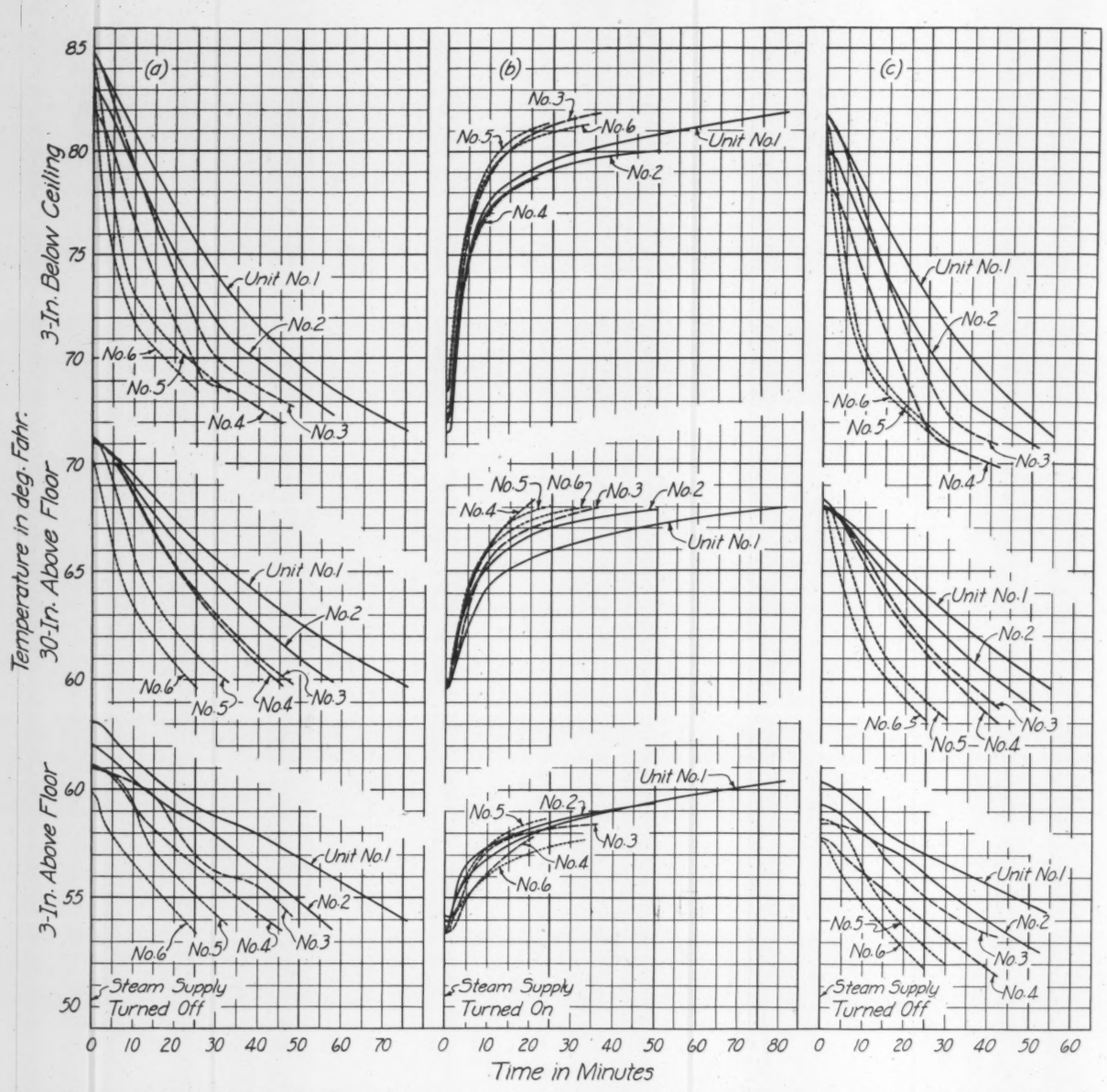

Fig. 8—Cooling and heating curves for room at 3-in. above floor, 30-in. above floor and 3-in. below ceiling

and 3 in. below the ceiling for the two cooling periods and the one heating period are shown in Fig. 8, while the equivalent temperatures and the average of the temperatures at the four levels in the room are included in Fig. 9. In every case the results obtained with the direct cast-iron radiators, units Nos. 1 and 2, are represented by full lines, those obtained with the convectors having cast-iron heating units, units Nos. 3 and 4, are representing by dashed lines, and those obtained with convectors having non-ferrous heating units, units Nos. 5 and 6, are represented by dotted lines.

Referring to Fig. 8a, which gives the results obtained during the first cooling period, it may be observed that with the exception of unit No. 6, the temperatures at the 30-in. level at the beginning of the period were practically the same with all of the units. Also, from this group of curves it may be noted that when the steam supply was turned off, the rate at which the room cooled at the 30-in. level was slightly different with the two convectors, units Nos. 5 and 6, which had non-ferrous heating units, and was practically the same with the two convectors, units Nos. 3 and 4, which had cast-iron heating units. The cooling rates with the two direct radiators were not the same, but they were both less than the rates obtained with any of the convectors. The upper and lower groups of curves in Fig. 8a give the temperatures near the ceiling and near the floor for all of the units tested, and it may be observed that in both of these groups of curves the relative position of each curve is the same as it is in the center group, which gives the temperature at the 30-in. level. Likewise, in Fig. 9a, which gives curves showing the equivalent temperature and the average of the temperatures of the air at four levels in the center of the room for each unit during the first cooling period, the magnitudes of the slopes of the curves for the different units are related to one another in the same way as those shown in Fig. 8a.

While all of the groups of curves in Figs. 8a and 9a show practically the same relationship between the rates of cooling with the different units, the group of curves

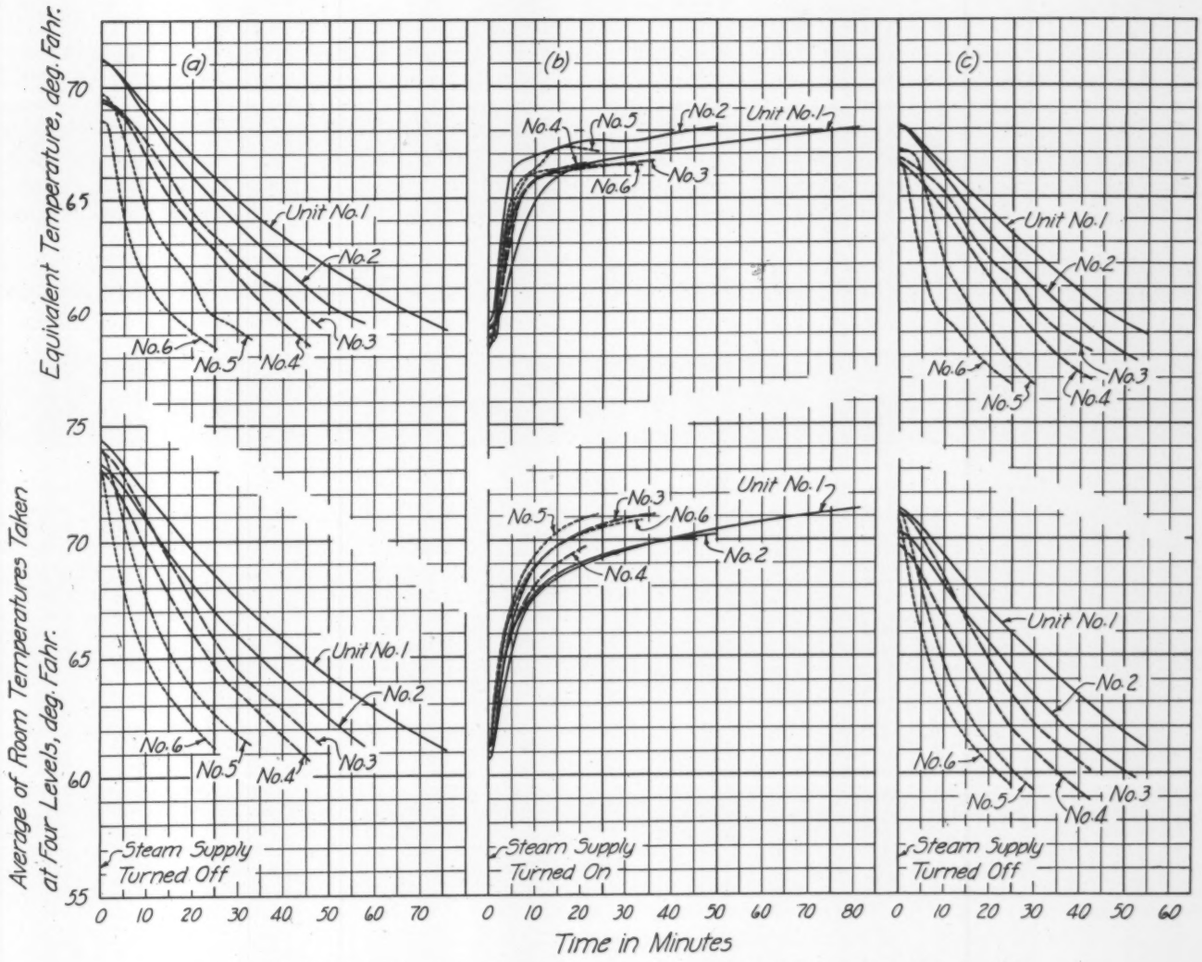

Fig. 9—Equivalent temperature and average of room temperatures taken at four levels

showing the average of the room temperatures at four levels has been selected for the purpose of correlating the cooling rates with the physical and thermal factors governing the performance of the units and the test room at the beginning of and during the first cooling period. The major factors affecting the rate at which a room cools when the steam supply to the heating unit is turned off may be divided into two general divisions, one pertaining to the room itself, and the other pertaining to the heating units. The first division would include such factors as the available thermal energy stored in the materials in the walls, floor, ceiling, furnishings, and in the air, and the rate at which heat is being lost from the air to the ambient spaces. Indices relating to the magnitude of these factors are the temperatures and masses of the different materials and the differences in temperature between the air in the room and that in the ambient spaces. In Fig. 9a it may be observed that with all of the units, the deviation in the averages of the temperature of the air in the test room at the start of the cooling period was less than 1.5 F. Referring to Fig. 4, under equilibrium conditions, it may be noted that the temperatures in the ambient spaces were approximately the same for all of the tests. Therefore, insofar as it was possible to obtain them, most of the influencing factors pertaining to the room itself were the same for each unit at the start of the first cooling period. Two appreciable exceptions to this similarity in conditions are indicated by the data in cols. 12 and 16, Table 2. Col. 12 gives the temperature of the inside surface of the exposed wall back of each unit under equilibrium conditions and just preceding the first cooling period. A comparison of these temperatures shows that the wall at this particular location was much warmer, and therefore, possessed more thermal energy at the start of the cooling periods with the radiators than it did at the start of the cooling periods with the convectors.

The difference between the average of the temperature of the air in the center of the room at the 30-in. and 60-in. levels, and 3 in. below the ceiling, and the average of the temperatures of the inside surfaces of the two exposed walls taken at the same levels as the air temperatures, are given in col. 16, Table 2. It may be noted from these data that the difference between the temperature of the air in the room and the temperature of the inside surfaces of the wall was always slightly less with the radiators than it was with the convectors. This indicated that for a given air temperature in the room, the temperature of the exposed walls and, therefore, the stored thermal energy, was slightly higher at the beginning of the cooling periods with the radiators than it was at the beginning of the cooling periods with the convectors. No doubt both of the conditions discussed result from the difference in the manner in which heat is emitted from radiators and convectors, and both of them would tend to cause the room to cool more

slowly with radiators than with convectors.

The factors pertaining to the units themselves which might influence the rate at which the room cooled consist of the available thermal energies in the masses of metal constituting the units and in the steam in the units at the time the steam supply was turned off. Another contributing factor might consist of the effectiveness of the unit in transmitting the available energy to the air in the room. A study of the first two factors involved the weight of each unit and the volume of the steam space, which are given in cols. 12 and 13 in Table 1. Unit No. 1, the radiator with the sections on 2½-in. centers, was the heaviest unit tested, and it also contained the largest volume of steam space. The weight of this unit was 151.0 lb and the volume of its steam space was 0.3180 cu ft. Unit No. 6, a convector having a non-ferrous heating unit, was the lightest unit tested, and it contained the smallest volume of steam space. The total weight of this convector, including the heating unit and the cabinet, was only 43.0 lb and the volume of its steam space was only 0.0162 cu ft. Thus the weight of the heaviest unit was about 3.5 times the weight of the lightest, while the ratio of the maximum steam space to the minimum steam space was 19.6. It is interesting to note the wide variation in the weight of the two convectors having non-ferrous heating units, and also to note that, with the cabinets included, units Nos. 3 and 4, the convectors with cast-iron heating units, both weighed slightly more than the direct radiator having the sections on 1½-in. centers. It is possible that the values in col. 23, Table 1, which give the total weight of metal in pounds per square foot of equivalent direct radiation (EDR) have greater significance than the total weight of the units. These values range from 1.81 to 5.37 lb per square foot of equivalent direct radiation (EDR), the minimum and maximum values being those for the lightest and heaviest units respectively.

As previously stated, the temperatures of the heating surfaces were observed by means of a thermocouple attached to a section of prime heating surface. In the case of the radiators in which the entire heating surface was prime surface, this temperature was for all practical purposes representative of the temperature of the entire mass of metal, and there was reasonable justification for its use in calculating the available energy stored in the unit. However, with the complex combinations of prime and secondary heating surfaces composing the heating units of the convectors, it was impossible to arrive at a temperature that was accurately representative of the actual mean temperature of the mass of material in the unit. Doubtless the temperature thus obtained on a portion of the prime heating surface was higher than the mean temperature of the material in the unit and its use in calculating the available energy resulted in values that were somewhat too large. Similarly, the determination of an accurate mean temperature for the metal in the cabinets of the convectors involved refinements that were regarded as unwarranted for the purposes of these tests. Therefore, the temperatures given in cols. 15 and 16, Table 1, while correct for the portion of the units where the thermocouples were attached, are, particularly in the case of the convectors, only approximations of the mean temperatures of the masses of metal in the entire units.

The approximate thermal energies available above 60 F in the metal in each unit and cabinet at the beginning of the first cooling period are given in cols. 24, 25, and 26, Table 1. These values were calculated by using the masses of material in the respective units, the specific heats of the metals, and the approximate temperatures given in cols. 15 and 16. They are, therefore, subject to the same limitations as the temperatures involved. The energy in the steam contained in each unit at the time the supply was turned off is given in col. 27, and the approximate total energy is shown in col. 28. These data show that while the quantities of energy contained in the steam in the different units varied over a range of from 15 to 1, the actual magnitudes of the quantities involved were so small in comparison with those contained in the metal of the units that the effect on the cooling rates of the room was probably negligible. Therefore, it appears that of the factors pertaining to the units themselves, the available energy stored in the metal of the heating units at the time the steam supply was turned off was the one which had the greatest effect on the rate at which the room cooled.

In correlating the different rates at which the room cooled with the different quantities of available energy in the various units at the time that the steam supply was turned off, the values given in col. 24, Table 1, are probably the most logical ones to use. These values include only the energy stored in the heating units and in using them the thermal energy stored in the cabinets of the convectors is neglected. This partly compensates for the fact that in the case of the convectors, the calculated available thermal energies stored in the heating units were somewhat too large, as previously explained. Referring to cols. 10 and 24, it may be noted that in each instance the magnitudes of the calculated available energies fall in the same relative order as the weights of the units. The curves in Fig. 9a which give the average of the temperatures of the air in the room at four levels, are probably the most representative of the conditions in the room at the beginning of and during the first cooling period. These curves exhibit the same characteristics as those for the 30-in. level in Fig. 8a. The latter, however, since they start at the same temperature, afford a more direct means for comparing the slopes of the individual curves. Differences in cooling rates may be observed by comparing these slopes. The curves for the two radiators show that the rate of cooling with unit No. 1, the heavier of the two radiators, was slightly less than it was for unit No. 2. The curves for the two convectors having cast-iron heating units indicate that there was practically no difference in the cooling rates with the two units, although the heating unit for unit No. 4 was slightly heavier than the one for unit No. 3. The curves for the two convectors having non-ferrous heating units show that the cooling rate with unit No. 6 was slightly greater than it was with unit No. 5, especially during the first portion of the cooling period. As the heating unit of unit No. 6 was considerably lighter than the heating unit of unit No. 5, the larger cooling rate was obtained with the lightest unit. On comparing the curves for units Nos. 3 and 4, with those for units Nos. 5 and 6, it may be seen that the cooling rates with the lighter types of convectors were greater than they were with the heavier types. Likewise, the cooling

rates with the radiators, which were heavier than any of the convectors, were less than the cooling rates with the convectors. Thus, it is apparent that the mass of metal in a heating unit, which is available for the storage of thermal energy, may become a material factor in affecting the cooling rate of a room when the steam supply is turned off. A comparison of the data given in cols. 20 and 33 of Table 1 still further confirms this conclusion.

Column 20 gives the heat emitted to the test room over a period of 10 min. under equilibrium conditions,

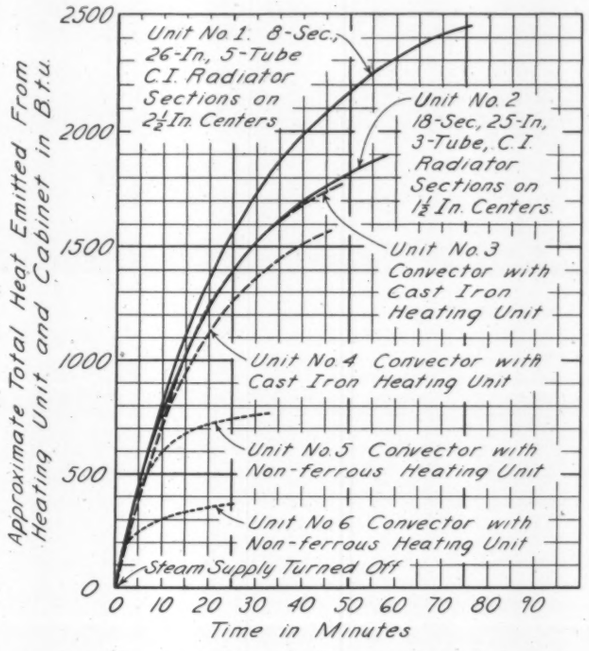
Approximate Total Heat Emitted From Heating Unit and Cabinet in B.t.u.
2500
2000
1500
1000
500
0
Unit No. 1. 8-Sec., 26-In., 5-Tube C.I. Radiator Sections on 2½ In. Centers
Unit No. 2 18-Sec., 25-In., 3-Tube, C.I. Radiator Sections on 1½ In. Centers
Unit No. 3 Convector with Cast Iron Heating Unit
Unit No. 4 Convector with Cast Iron Heating Unit
Unit No. 5 Convector with Non-ferrous Heating Unit
Unit No. 6 Convector with Non-ferrous Heating Unit
Steam Supply Turned Off
0 10 20 30 40 50 60 70 80 90
Time in Minutes

Fig. 10—Approximate heat emitted from heating units and cabinets after steam supply was turned off

and since equilibrium exists during this time, these values also represent the heat loss from the room. Column 33 gives the approximate heat emitted to the room from the various units, during the 10 min. period following the turning off of the steam supply, and a comparison of these quantities with those in col. 20 shows that, particularly in the case of the heavier units, the quantities of heat emitted from the metal in the units during the first part of the cooling period were of sufficient magnitude to tend to maintain the air temperature in the room. Also, the differences in the quantities of heat emitted from the light and heavy units were large enough to make it reasonable to expect a variation in the cooling rates. Fig. 10 shows the approximate heat emitted from the units at various time intervals during the entire cooling period. In the case of the convectors the approximate heat emitted from the cabinet is included with that emitted from the heating units. The curves were obtained from calculations based on the weights of the heating units and cabinets, the specific heats of the metals, and the temperatures previously discussed. They are, therefore, subject to the limitations inherent in the temperature observations, which, as previously explained, in the case of convectors, were somewhat higher than the actual mean temperature of all of the metal in the entire heating units and cabinets. A comparison of the curves for the different units, subsequent to the elapse of 15 min of the cooling period, shows that with one exception there was an appreciable difference in the approximate quantity of thermal energy that had been emitted from the units to the air and surroundings. This one exception was with units Nos. 2 and 3, and it is probable that, since the method of calculation gave results somewhat too great for all of the convectors, the curve for the heaviest convector would overlap that for the lightest radiator which consisted entirely of prime surface and had no cabinet. Furthermore, the difference in weight between units Nos. 2 and 3 was comparatively small.

In Fig. 9c the curves show the equivalent temperature and the average of the temperature of the air in the room at four levels for all of the units during the second cooling period. This cooling period started at the end of the warming period shown in Fig. 9b. These curves exhibited the same characteristics as those shown in Fig. 9a and hence lead to the same conclusions. In general, the results obtained during the two cooling periods indicated that the mass of material and the thermal energy stored in the units at the beginning of the periods had an appreciable effect on the cooling rates. However, any attempt to calculate the temperature of the air in the room or the temperature of the heating unit by the use of equations based on the assumption of equilibrium between the heat emitted from the unit and the instantaneous heat loss from the room indicated that the influence of the heat capacity of the walls, floor, and ceiling of the room was sufficiently great to invalidate the results obtained by such calculations. The calculated temperatures of both the air in the room and of the metal in the heating units gave cooling rates that were much greater than those actually obtained by test.

The results obtained during the heating period are shown in Figs. 8b and 9b. The center group of curves in Fig. 8b shows the rates at which the temperature of the air at the 30-in. level increased when the steam supply was turned on, and the lower and upper groups of curves give the same information with respect to the temperature of the air 3 in. above the floor and 3 in. below the ceiling. From the data in Fig. 8b it may be observed that with each unit the temperature of the air near the ceiling increased at a greater rate than that for the temperature of the air at the 30-in. level. Also, in every case the rate of heating of the air at the 30-in. level was greater than the rate of heating of the air 3 in. above the floor. Referring to the center group of curves in Fig. 8a, it may be noted that under equilibrium conditions, or at the beginning of the cooling period, all of the units, except unit No. 6, were of sufficient size to maintain practically the same temperature at the 30-in. level.

From the center group of curves in Fig. 8b it may be observed that during the first 8 min of the heating period the heating rates with all of the units, with the exception of unit No. 1, were practically the same. From that time on, the rates with units Nos. 2 and 3 decreased, and during the latter part of the period they became considerably less than the rates with units Nos. 4 and 5. These latter two units, one a convector with

a cast-iron heating unit, and one a convector with a non-ferrous heating unit, maintained approximately the same heating rates during the greater portion of the heating period. The heating rate with the heaviest radiator, unit No. 1, was less than the heating rate with all of the other units during the entire period, and that for unit No. 2, the light weight radiator, was less than those with the convectors, after about 8 min of the period had elapsed. While the results based on the rate of heating of the air at the 30-in. level indicated that the heating rates with the radiators were less than those with the convectors, they did not show any consistent difference between the heating rates with the convectors having cast-iron heating units, and those having non-ferrous heating units.

The performances of the units during the heating period may also be compared on the basis of the average of the temperatures of the air at four levels in the room. These data are given in Fig. 9b, and from them it may be observed that on this basis the heating rates with the two radiators were approximately the same. The heating rates with all of the convectors were greater than those with the radiators, and although the difference between the rates with the various convectors was not great, the results indicated that the rates with the convectors having non-ferrous heating units were somewhat greater than those with the convectors having cast-iron heating units.

In general, the differences between the performances of the various units during the heating period were not as great nor as clearly marked as they were during the cooling periods. This, no doubt, was due to the fact that, within 5 min after the steam was turned on at the start of the heating period, the air in the steam space was all expelled and the heating surfaces attained full operating temperatures. From that time on the heating of the room was independent of the masses of the units, and was dependent upon the capacities of the units and the effectiveness with which they emitted heat to the air and the surroundings. Inasmuch as the capacities of the convectors were practically the same, any small differences in the rates at which the room heated would be explained by slight differences in the heating effect of the individual convectors. The heating effect of the radiators differed from that of the convectors in that a much larger portion of the heat was given off by direct radiation. This direct radiation heated the walls and surroundings without having an immediate effect on the temperature of the air in the room. Hence, the rate of heating the room, as measured by the rate of temperature rise for the air, was somewhat less for the radiators than it was for the convectors. That considerable effective radiation was present in the room, particularly in the case with the radiator designated as unit No. 2, is shown by the upper group of curves in Fig. 9b. These curves give the equivalent temperatures in the room, as observed with the eupatheoscope, during the heating periods with all of the units. As previously explained, the eupatheoscope evaluates the conditions in a non-uniform environment in terms of the temperature in a uniform environment, taking into account the presence of radiation from hot and cold surfaces. The effectiveness of radiation from the hot surfaces of unit No. 2 on the equivalent temperature, may be observed by comparing the relative position of the curves for unit No. 2 in the two groups of curves in Fig. 9b. From these curves it is evident that while the average of the temperature of the air at four levels in the room with unit No. 2 was lower than it was with the convectors, the equivalent temperature with the radiator was higher than it was for any of the convectors.

Summary and Conclusions

The results of this investigation indicate that considerable differences may exist between the cooling or heating rates of a room as influenced by different types of heating units. Such results may be useful in the application to problems of temperature control, and particularly to problems arising from the use of different types of heating units in the same heating system.

The following conclusions may be drawn as applying to the conditions under which the tests were conducted:

1. Under equilibrium conditions corresponding with those in winter service, a room can be heated to the same temperature at the 30-in. level with less steam when the latter is used in convectors than when it is used in direct radiators. However, the equivalent temperature produced by convectors is from 1.3 to 2.2 F lower than that produced by direct radiators and any saving in steam consumption is in some measure offset by less comfortable conditions in the room.

2. The rate of cooling of a room is materially influenced by the mass of the heating unit and hence by the thermal energy stored in the unit at the time that the steam is turned off. The lower rates of cooling correspond with the greater masses of the heating units. For the units used in these tests the rate of cooling of the room was greatest with the convectors having non-ferrous heating units, was somewhat less with convectors having cast-iron heating units, and was least with direct radiators.

3. The rate of cooling of a room is greatly influenced by the heat capacity of the materials in the walls, floor, and ceiling. The evaluation of this effect is very difficult and uncertain.

4. The thermal energy of the steam in the unit at the time that the steam is turned off is comparatively small and has no effect on the rate of cooling of the room.

5. Under the venting conditions maintained on these tests, the air in the steam space was expelled and the heating surfaces attained full operating temperatures within 5 min after the steam was turned on. Under such conditions the rate at which the room heats is practically independent of the masses of the heating units, but is dependent on the capacities of the units and the effectiveness with which they emit heat to the air and the surroundings.

Acknowledgments

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which M. L. Enger, Dean of the College of Engineering, is the Director. The investigation was conducted in the Department of Mechanical Engineering of which O. A. Leutwiler, Professor of Mechanical Engineering Design, is the Head. The results will ultimately comprise part of a bulletin of the Engineering Experiment Station.

Summer and Winter Air Conditioning by Low Pressure Steam

By G. F. Zellhoefer* (*NON-MEMBER*), Bloomington, Illinois

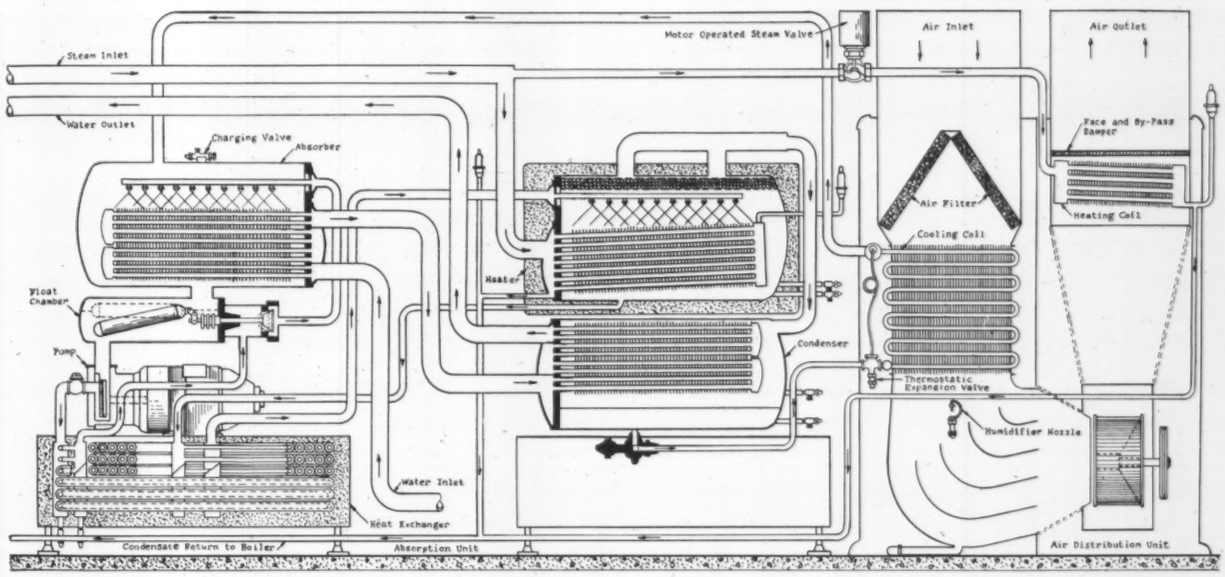

Fig. 1—Operating cycle of absorption unit

THERE has been developed a complete air conditioning plant in which cooling is supplied through a new type of absorption refrigeration unit operated on low pressure steam. This air conditioning plant is designed as an integral unit in sizes ranging from 3 to 25 ton cooling capacity and 75,000 to 750,000 Btu per hour heating capacity. The equipment for heating, humidification, forced air circulation, ventilation and air cleaning is designed in accordance with conventional practice but the refrigeration unit is an entirely new development. The purpose of this paper is to discuss this new absorption refrigeration unit and its adaptation to the complete air conditioning plant.

Cycle of Operation

A schematic drawing of the unit to illustrate the cycle of operation is shown in Fig. 1. The solution of solvent and refrigerant is sprayed over a steam heated coil in the heater. The refrigerant is distilled from the solution and this gaseous refrigerant passes into the condenser where it is chilled and condensed to a liquid. This liquid refrigerant passes from the condenser through a thermal expansion valve into the cooling coils where it is vaporized and the gaseous refrigerant then passes into the absorber where it is absorbed by the weak solution. While the refrigerant is passing through this cycle of operation, the weak solution formed by the distillation of the refrigerant passes from the heater through the solution heat exchanger to the absorber, where it dissolves the gaseous refrigerant coming from the cooling coil. The solution of refrigerant and solvent is pumped through the solution heat exchanger and is sprayed into the heater, thereby completing the cycle of operation.

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Characteristics of the Chemicals

The solvent is dimethyl ether of tetraethylene glycol $CH_3\ (OCH_2\ CH_2)_4\ OCH_3$ and the refrigerant is dichloromonofluoromethane ($CHCL_2F$). Both chemicals are essentially non-toxic, non-irritant, non-corrosive, non-inflammable, chemically stable, chemically inert and have a low specific heat with a high coefficient of heat conductivity.

The curve for latent heat of vaporization of the refrigerant is given in Fig. 2. The vapor pressure curve is given in Fig. 3. The specific gravity of the liquid at 75 F is 1.47. The specific heat of the liquid is 0.258 and the c_p (specific heat at constant pressure) of the vapors is 0.129—0.0000886t in Btu per pound, t in degrees Fahrenheit. The latent heat of vaporization, the solubility characteristics, and the vapor pressure of dichloromonofluoromethane differ greatly from that of dichlorodifluoromethane, but the other chemical and physical properties of the chemicals are similar.

The boiling point of the solvent is 505 to 511 F at atmospheric pressure. The vapor pressure curve of the solvent at 232 F to 265 F is given in Fig. 4. The

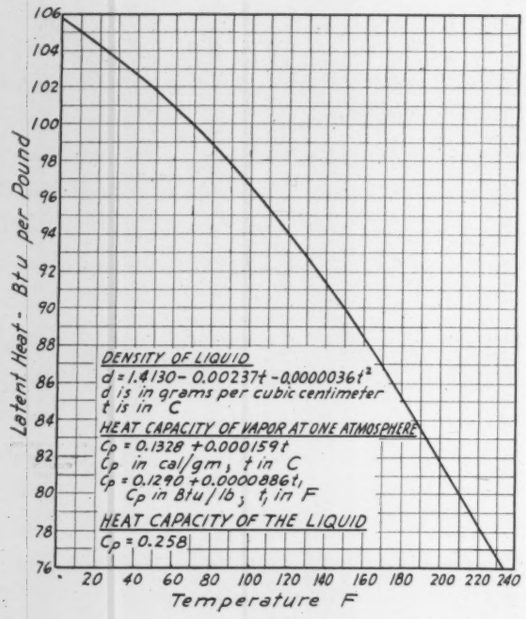

Latent Heat - Btu per Pound
Temperature F
DENSITY OF LIQUID
d = 1.4130 - 0.00237t - 0.0000036t²
d is in grams per cubic centimeter
t is in C
HEAT CAPACITY OF VAPOR AT ONE ATMOSPHERE
Cp = 0.1328 + 0.000159t
Cp in cal/gm, t in C
Cp = 0.1290 + 0.0000886t,
Cp in Btu/lb, t, in F
HEAT CAPACITY OF THE LIQUID
Cp = 0.258

Fig. 2—Latent heat of vaporization curve for dichloromonofluoromethane

specific gravity at 75 F is 1.007 and the specific heat at 75 F is 0.513.

The solubility curve of the refrigerant and the solvent under the operating conditions of the absorber is given in Fig. 5, while the solubility characteristics under the operating conditions of the heater is given in Fig. 6.

Characteristics of Performance

The absorption refrigeration units are rated under the following operating conditions: 85 F inlet cooling water, 95.5 F outlet cooling water, 10 lb steam pressure and 40 F saturated refrigerant temperature in the cooling coils. The characteristics of a unit operating under these conditions is given in Table 1.

From the graph in Fig. 7 may be determined the steam requirements and the capacities of a nominal 5½ ton unit through a wide range of operating conditions. The dotted line indicates the capacities and steam requirements at the operating conditions specified for nominal rating. Under these conditions, this unit has a capacity of 5.55 tons and the steam requirements are 130 lb per hour or 23.4 lb per hour per ton. The heat in condensing 23.4 lb of steam at 10 lb pressure is 22,230 Btu. The thermal efficiency of this unit expressed as the ratio of the refrigeration effect to the heat supplied by the steam at these operating conditions is 54 per cent. Efficiencies as high as 62 per cent may be obtained by using a larger solution heat exchanger, but for most installations, the investment in the larger heat exchanger is not warranted by the increase in thermal efficiency.

Through this 5½ ton unit there was circulated 36 gal of cooling water per minute. At this rate of flow, there was a rise of 10.5 F when the unit developed its rated capacity of 5.55 tons. In other operating conditions indicated on the graph in Fig. 8, the volume of cooling water flowing through the unit remained constant, and the temperature rise varied with the load.

The heat dissipated in the water is equivalent to the sum of the heat liberated in the condensation of the steam and the heat absorbed by the vaporization of the refrigerant in the cooling coil. This is 565 Btu per minute per ton of refrigeration produced at the operating conditions specified for the nominal rating of the unit, but varies slightly with operating conditions.

The turbine type pump is used to circulate the strong liquor from the absorber through the heat exchanger to the heater. As indicated in Table 1, the hydraulic horsepower is 0.036 and the total horsepower 0.09 per ton

Table 1—Operating Performance Characteristics of Absorption Refrigeration Unit

Cooling water entering absorber	85	F
Cooling water leaving absorber	91.25	F
Cooling water entering condenser	91.25	F
Cooling water leaving condenser	95.50	F
Volume of cooling water circulated per minute per ton	6.5	gal
Temperature rise in cooling water passing through the unit	10.5	F
Heat dissipated in the cooling water per minute per ton	565	Btu
Pressure drop of water in passing through the unit	10	lb
Weak liquor temperature entering absorber	125	F
Strong liquor temperature leaving absorber	95	F
Weak liquor temperature leaving heater	232	F
Strong liquor temperature entering heater	188	F
Liquid refrigerant leaving condenser	98	F
Condenser pressure	28	lb
Low side pressure, vacuum	5	in. of Hg.
Pressure at pump outlet	58	lb
Pressure at heater nozzles	36	lb
Pressure on absorber float valve	5	lb
Pressure on absorber nozzles	5	lb gage
Total pressure on absorber nozzles	7.5	lb
Steam required per ton per hour at 10 lb pressure (22,230 Btu per hour)	23.4	lb
Theoretical hydraulic horsepower required for circulating the strong liquor per ton	0.036	hp
Total horsepower required for circulating the strong liquor per ton	0.09	hp
Circulation of weak liquor per minute per ton (theory)	0.48	gal
Circulation of weak liquor per minute per ton (actual)	0.64	gal
Circulation of strong liquor per minute per ton (theory)	0.68	gal
Circulation of strong liquor per minute per ton (actual)	0.90	gal
Circulation of refrigerant per minute per ton	2.25	lb
Prime surface on the heater coil per ton of refrigeration capacity	3.4	sq ft
Extended surface on heater coil per ton of refrigeration capacity	14.00	sq ft
Prime surface on the condenser coil per ton of refrigeration capacity	4.00	sq ft
Extended surface on condenser coil per ton of refrigeration capacity	16.00	sq ft
Prime surface on absorber coil per ton of refrigeration capacity	4.00	sq ft
Extended surface on absorber coil per ton of refrigeration capacity	15.00	sq ft
Surface on inner tube of heat exchanger per ton of refrigeration	1.9	sq ft
Approximate weight of unit per ton capacity	180	lb

of refrigeration at operating conditions specified for the nominal rating of the unit. This horsepower requirement for a unit remains essentially constant regardless of operating conditions and the corresponding refrigeration capacity.

The graph shown in Fig. 8 indicates the theoretical volume of weak and strong liquors circulated per minute per ton with a saturated refrigerant temperature of 40 F in the cooling coil, 28 lb head pressure, absorber temperatures within the range of 85 to 110 F and the heater temperature within the range of 230 and 245 F. Under the operating conditions specified for nominal rating of the unit, the solution leaving the absorber is 76 per cent of complete saturation, and the solution leaving the heater is supersaturated 2 per cent. Consequently, the actual rate of circulation is 33⅓ per cent greater than theory. This per cent saturation varies slightly with operating conditions.

Under the operating conditions specified for the nominal rating of the unit, the temperature of the weak liquor leaving the heater is 232 F, and the pressure in the heater is 28 lb gage. Based on this temperature, the vapor pressure of the solvent is 1 mm, and the partial vapor pressure at 28 lb gage is 0.05 per cent of the total pressure. In a mixture of the solvent and refrigerant vapors at this temperature and pressure the per cent by weight of solvent vapors is theoretically 0.11 per cent. In practice, it is found that percentage by weight of solvent carried into the rectifier is approximately 0.15 per cent or 0.035 lb per minute per ton. The heat loss due to the carry-over of the solvent is less than 3 per cent of the total heat input.

From Table 1 may be determined the required size of each part of the absorption refrigeration unit. The overall dimensions of a unit depend on design. In the case of a 20 ton unit, the overall dimensions are 48x48x67 in. high, including the condenser. Line drawings of such a unit are shown in Fig. 9. In the type of unit shown in Fig. 9, cooling water is circulated through the absorber and condenser. In some types of installation, it is an advantage to cool the absorber and condenser by evaporative cooling. Such units are designed with the absorber refrigeration unit as an integral part with the water-cooling equipment. A schematic drawing of such a unit is shown in Fig. 10.

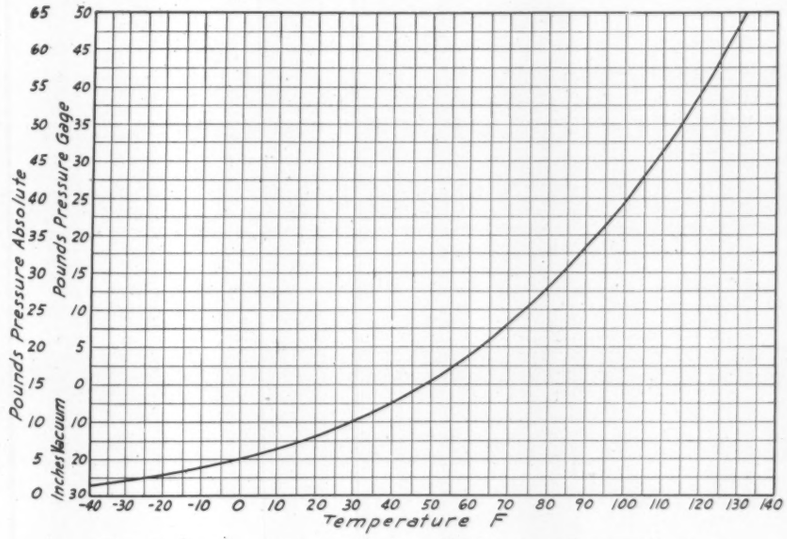

Pounds Pressure Absolute
Pounds Pressure Gage
Inches Vacuum
Temperature F
-40 -30 -20 -10 0 10 20 30 40 50 60 70 80 90 100 110 120 130 140

Fig. 3—Vapor pressure curve for dichloromonofluoromethane

Method of Testing

In making the test on a unit of this type, the refrigeration effect of the machine is dissipated in an electrically heated water bath maintained at room temperature. The current required to maintain the bath temperature is recorded by an integrating meter. The refrigeration effect due to super-heating the vapors in excess of 10 F is calculated and this amount subtracted from the total heat input of electric heaters to obtain the net refrigeration effect of the machine.

The steam used in the heater is passed through an eliminator to remove any entrained moisture. The pressure and temperature of the steam are taken at the heater inlet. The quantity of steam condensate from the heater is determined by

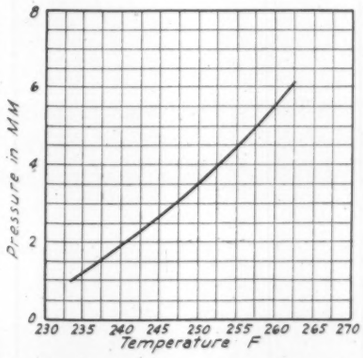


Fig. 4—Vapor pressure curve for dimethyl ether of tetraethylene glycol

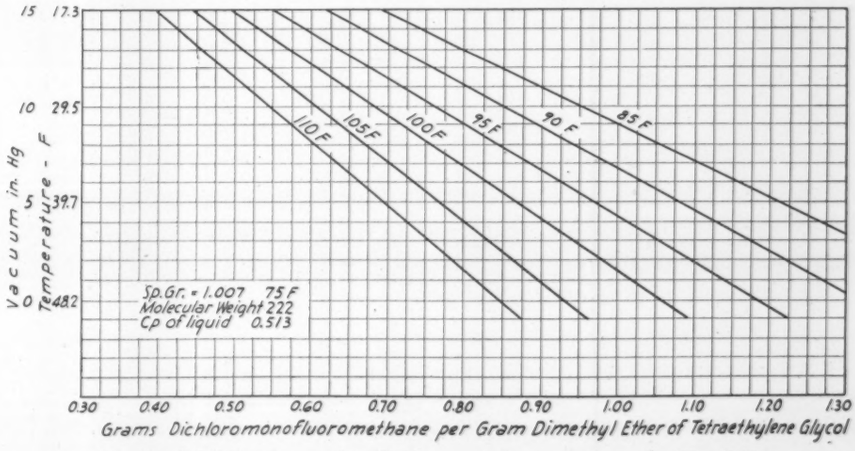


Fig. 5—Solubility curve for dichloromonofluoromethane in dimethyl ether of tetraethylene glycol under operating conditions of absorber

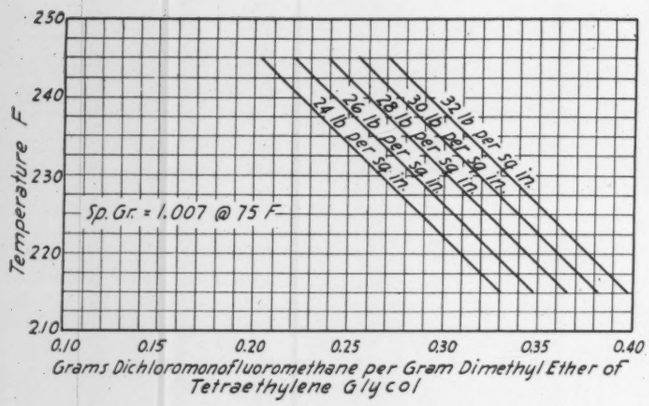


Fig. 6—Solubility curve for dichloromonofluoromethane in dimethyl ether of tetraethylene glycol under operating conditions of heater

allowing the condensate to flow by gravity through a float type steam trap and then through a water cooled coil to a receiver, where it is weighed. The heat input is determined by calculating the heat liberated in the condensation of the steam at the pressure prevailing in the heater.

The concentration of refrigerant in the strong solution is determined as follows: A 50 cc sample of strong solution is drained from the absorber into an evacuated 100 cc container submerged in ice water. The weight of the solution is determined and then the solution is refluxed at 510 F to distill off the refrigerant. The remaining solvent is weighed and the per cent concentration of refrigerant in the sample of solution calculated. This concentration is approximately 75 per cent of the complete saturation for the temperature and pressure prevailing in the absorber. The solubility at complete saturation for the range of conditions prevailing in the absorber is indicated in Fig. 5.

In like manner, a sample of weak solution is drawn from the outlet of the heat exchanger and the concentration of refrigerant determined. The solvent is found to be supersaturated approximately 2 per cent as it leaves the heater. Concentration for the range of conditions prevailing in the heater is given in Fig. 6.

The amount of solvent carried over from the heater into the receiver is determined as follows: A 100 cc sample of liquid refrigerant is discharged from the receiver through a small coil submerged in ice water into 100 cc centrifuge tube which also is submerged in ice water. The refrigerant is allowed to distill from this tube at room temperature until the volume is reduced to approximately 3 cc and then it is slowly heated to 100 F and the tube is evacuated to a vacuum 26 in. Hg. The solution remaining is then saturated with refrigerant vapors under a pressure of 25 in. of Hg absolute and at 95 F. At this temperature and pressure this solution is by weight 50 per cent solvent and 50 per cent refrigerant. The volume of this solution in cc multiplied by 0.6 is the per cent by weight of the refrigerant present in the original sample. Under standard operating conditions approximately 0.15 per cent of solvent is held in solution with the liquid refrigerant in the receiver.

The rate of flow of cooling water through the machine is determined by the pressure drop of the water in passing through a calibrated orifice. The machines are designed for a rate of flow of 6.5 gpm per ton. The temperature rise of the water in passing through the unit is determined by the use of glass mercury thermometers. These thermometers are placed in thermometer wells and the bulbs of the thermometers are submerged in oil. The temperature rise under standard operating conditions is 10.5 F. In the test work on a unit, the rate of flow was maintained constant for a given unit and the temperature rise allowed to vary with the load which depends upon operating conditions.

Units for field tests were subjected to the tests previously indicated prior to their installation. In the field the running time of the units was clocked and separate integrating meters were installed to determine power input. A record was kept of the fuel required to supply the unit with steam. The results of the field tests correspond with the laboratory data given in Table 1.

As part of the field tests, there was taken at regular intervals a sample of solution from several of the machines and a chemical analysis made on these samples to determine if there was any chemical reaction taking place within the unit. There was no evidence of reaction other than a slight hydrolysis of the refrigerant and the reaction of the acid so formed with the metals. This reaction was not sufficient to produce any perceptible corrosion.

Complete Air Conditioning Plant

The complete standard air conditioning plant consists of a boiler-burner unit, air distribution unit, absorption machine, and control system.

The boiler of a standard boiler-burner unit is especially designed for use with oil, and the burner is a low pressure atomization, gun-type burner. This unit is not an integral part of the rest of the equipment and any source of low pressure steam may be used with this air conditioning plant.

In the air distribution unit there are cooling coils, indirect steam radiation coils, filters, a humidifier and a motor-blower unit. Conventional practice is followed in the design of this unit, except for the cooling coils which are especially designed for use with dichloromonofluoromethane.

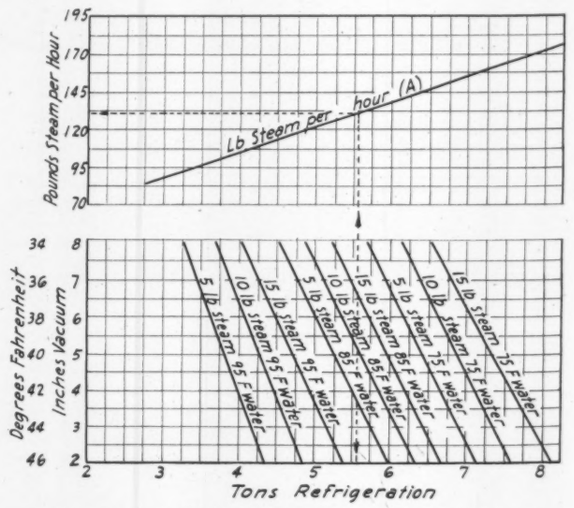


Fig. 7—Capacity and steam requirements of a test unit

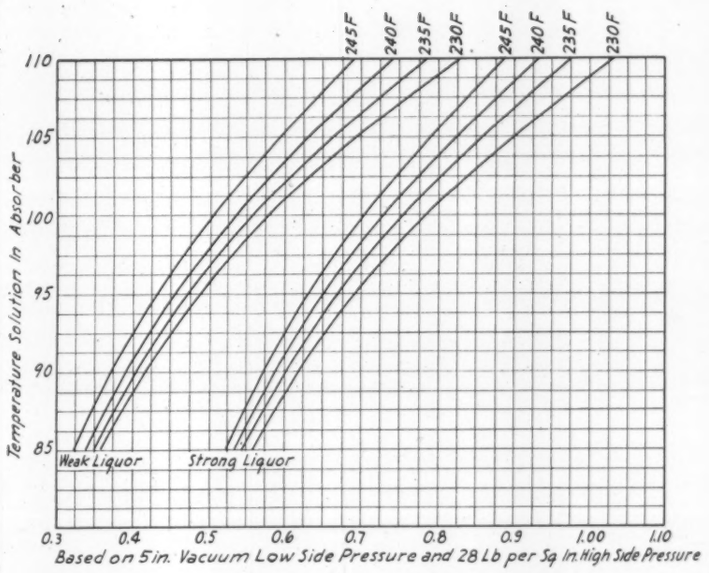


Fig. 8—Rate of circulation in gallons per minute per ton based on 100 per cent saturation of solution from absorber and heater

The control system consists of a selector switch, cooling thermostat, two heating thermostats, damper controller, humidistat and the necessary relays. The selector switch has five positions: *off*, *day heating*, *night heating*, *ventilation* and *cooling*. In the *off* position, the circuit is open to all equipment except the boiler-burner unit, which operates independently of the rest of the system by means of the pressure control. When the selector switch is turned to the *day heating* position, the motor-operated steam valve to the indirect steam radiation is opened, the circuit to the *day heating* thermostat is closed, and the circuit to the motor-blower unit is closed.

This thermostat then controls the temperature by controlling the position of the face and by-pass dampers over the indirect steam radiation coils. The blower operates continuously to provide continuous forced air circulation and ventilation. When the selector switch is set to the *night heating* position, the steam valve to the heating coils is opened, the circuit to the *night heating* thermostat is closed, the face dampers over the heating coils are opened and the by-pass dampers are closed. The *night thermostat* then controls the temperature by intermittent operation of the blower. On either of the *heating* positions, the blower operates at *low* speed.

On the *ventilation* position of the selector switch, the steam valve to the heating coils is closed and the circuit to the motor-blower unit is closed, thereby providing continuous forced air circulation and ventilation without heating or cooling. On the *ventilation* position of the selector switch, the blower operates at *high* speed.

When the selector switch is set to the *cooling* position, the circuit is closed to the cooling thermostat and motor-blower unit. The motor-operated steam valve to the indirect steam radiation is closed. The absorption refrigeration unit operates under control of the *cooling* thermostat and the blower operates continuously. On the *refrigeration* position of the selector switch, the blower operates at *high* speed.

The humidistat operates entirely independent of the rest of the system. It controls the humidity in the heating season by regulating the intermittent flow of water through a solenoid valve to the humidifier nozzles.

Solvents and Refrigerants

In the development of this absorption refrigeration unit an extensive study has been made of organic solvents for the chemically stable halogenated hydrocarbon refrigerants.

Chemically inert and stable organic liquids that have a low vapor pressure are limited to a relatively few classes of compounds. Compounds from each such class have been tested for their solubility characteristics with these refrigerants. In those classes of chemicals that proved to be good solvents for one or more of the refrigerants many new compounds have been synthesized in search of a chemical that possesses high solubility characteristics together with the other desirable characteristics for a solvent in an absorption refrigeration unit. An extensive study has been made of the following groups of chemicals:

RCH_2CH_2R'
$RCH_2CH_2OCH_2CH_2R'$
$RCH_2CH_2OCH_2CH_2OCH_2CH_2R'$
$RCH_2CH_2OCH_2CH_2OCH_2CH_2OCH_2CH_2R'$
$RCH_2CH_2OCH_2CH_2OCH_2CH_2OCH_2CH_2OCH_2CH_2OCH_2CH_2R'$
$RCH_2CH_2CH_2R'$
$(C_2H_7O)CH_2R$

Where R and R' are selected from the radicals:

OCH_3
OC_2H_5
$OCOCH_3$
$OCOCH_2OCH_3$
$OCOCH_2OC_2H_5$
$(C_2H_7O)CH_2O$

Many other types of ethers, esters and ether-ester compounds have been made and tested. A number of such compounds proved to be good solvents for one or more of the refrigerants.

There has been determined the solubility characteristics of a wide variety of aliphatic and aromatic halogenated hydrocarbons with the halogenated hydrocarbon refrigerants. Those chemicals in this category that were tested and found to have good solvent characteristics had other chemical or physical properties that mitigated their value for use in an absorption refrigeration unit. In general, high boiling alcohols and hydrocarbons are poor solvents for these refrigerants.

For each combination of solvent and refrigerant studied there has been calculated the required rate of circulation of strong liquors to produce one ton of refrigeration under the operating conditions specified for nominal rating of this absorption refrigeration unit. This required rate of circulation depends on the solubility characteristics, specific gravity of both chemicals, and the latent heat of vaporization of the refrigerant.

On the basis of the required rate of circulation of solution, these polyethylene glycol derivatives are better solvents in general for dichloromonofluoromethane,

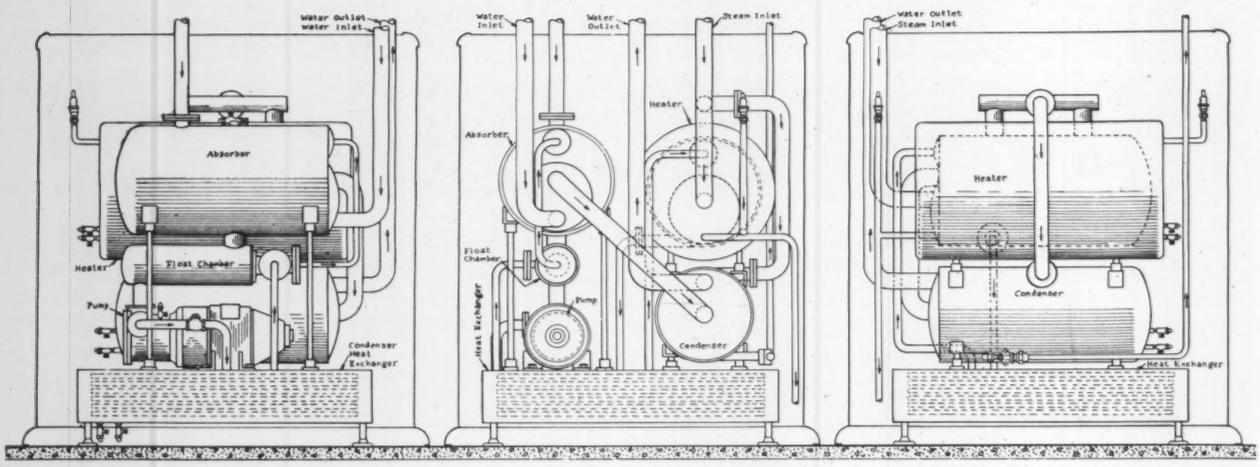


Fig. 9—Diagram of absorption refrigeration unit

monofluoromonochloromethane and difluoromonochloromethane than for other stable halogenated hydrocarbon refrigerants. In general, methyl chloride, ethyl chloride and methylene chloride compare more favorably than dichlorodifluoromethane, dichlorotetrafluoromethane, trichlorotrifluoromethane and trichloromonofluoromethane. In the polyethylene glycol derivatives, $RO(CH_2CH_2O)_xR$, the solubility of any given halogenated hydrocarbon refrigerant is not affected greatly by an increase in the value $_x$ but is affected by the nature of the R groups. In general, where these R groups are methyl, ethyl, acetyl, methoxyacetyl or ethoxyacetyl, better solubility is obtained than where the R groups contain a greater portion of hydrogen and carbon.

In addition to the required rate of circulation of the solution, other factors to be considered in developing a solvent-refrigerant combination are heat transfer characteristics, specific heats, vapor pressures, chemical stability, toxicity, inflammability and chemical inertness. Taking into consideration all these factors, the combination of dichloromonofluoromethane as refrigerant and dimethyl ether of tetraethylene glycol as solvent is one of the best combinations developed in this study for use in an absorption unit designed to operate under the conditions required in air conditioning.

Summary

There has been developed a complete air conditioning plant using a new type of absorption refrigeration unit. This refrigeration unit operates on low pressure steam with a thermal efficiency of 54 per cent.

An extensive study has been made of organic solvents for the chemically stable halogenated hydrocarbon refrigerants. In a refrigeration unit used for air conditioning, one of the best solvent-refrigerant combinations developed is dichloromonofluoromethane as refrigerant and dimethyl ether of tetraethylene glycol as solvent.

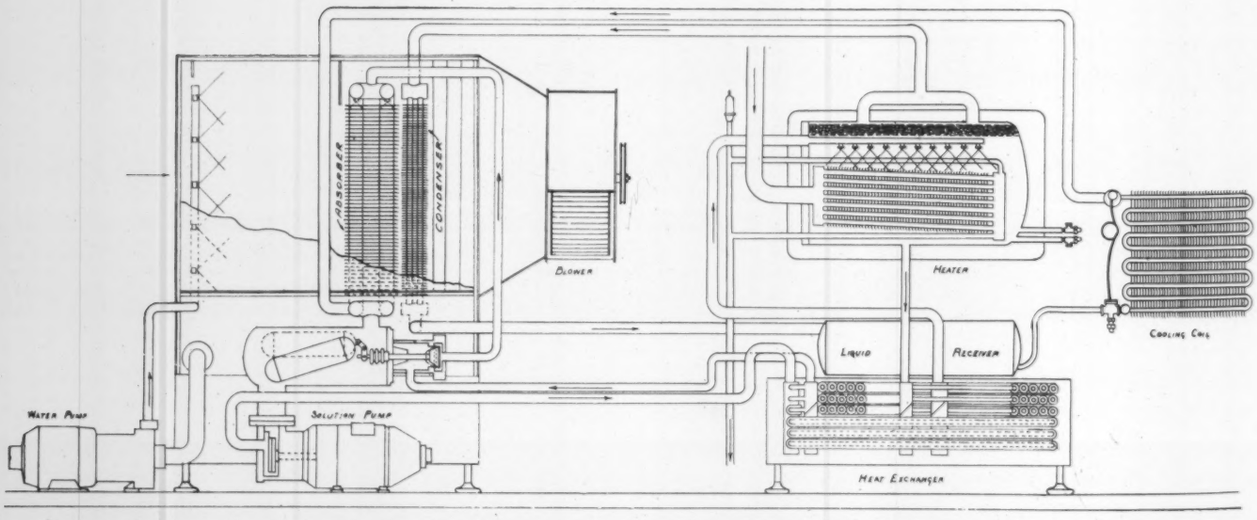


Fig. 10—Diagram of complete air conditioning unit

The Inspiration of Society Membership

Evidence of the value of membership in the Society is illustrated by the number of men who have been on the rolls for 25, 30 and 40 years, and from time to time during this year the Membership Committee will present the comments of some of these men on such topics as Why I Joined the Society; Why I Am Interested in the Society; and What I Value in Society Membership.

The first message in reply to the inquiry of R. C. Bolsinger, chairman of the Membership Committee, is given by W. H. Driscoll, Jersey City, N. J., who was president of the Society in 1926.

W. H. Driscoll

You have asked me why I have continued my interest and activity in the Society long after I have held the highest honors the Society can bestow and to that question I am happy to respond with more enthusiasm than it is possible to convey in this message.

It is my privilege to hold very happy memories of my contacts with the many great and noble characters whose spirits hover over our Society and exert a mighty influence on its affairs.

These men served the Society in days when there was little to inspire them beyond their own unselfish devotion to an ideal. Willingly and generously out of their own experience and from their personal data books, laboriously compiled, they offered all they had and all they knew of the arts of heating and ventilating at a time when the rule of thumb was an accepted formula, when knowledge and understanding of these subjects were personal and not general and when worthwhile text books on the subjects were rare.

There was Carpenter, and Kent, and Barron, and Jellett—there was Mackay, and Kinealy, and Oh! so many others whose words of wisdom poured forth eloquently while I sat in humbleness and awe and marveled at the greatness of their minds and the wealth of their experiences.

It was men of that calibre who worked so diligently to prepare the foundation on which our great Society was reared and every one of them remained active and interested members until death overtook them.

But though time takes its inevitable toll of all men, the Society has always been blessed with members of great ability and splendid character whose very presence at our meetings has amply repaid us for our own attendance.

There never has been a time in the history of our Society when its membership has not included an Allen or an Anderson, a Hill or a Harding, a Lewis or a Lyle, a Willard or a Carrier, or an endless number of others whose activities at our meetings have been a source of inspiration to all who attend.

Added to these there is that brilliant array of younger men constantly coming up with new and advanced ideas and carrying the standards of the profession to heights undreamed of at the inception of the Society.

I never attend a meeting that I do not carry away new and useful ideas and a finer appreciation of the Society membership. It is for these reasons that I attend meetings of the Society and the Chapter. If I contribute anything, it is merely incidental to the purpose of my visits.

Semi-Annual Meeting—June 24-26

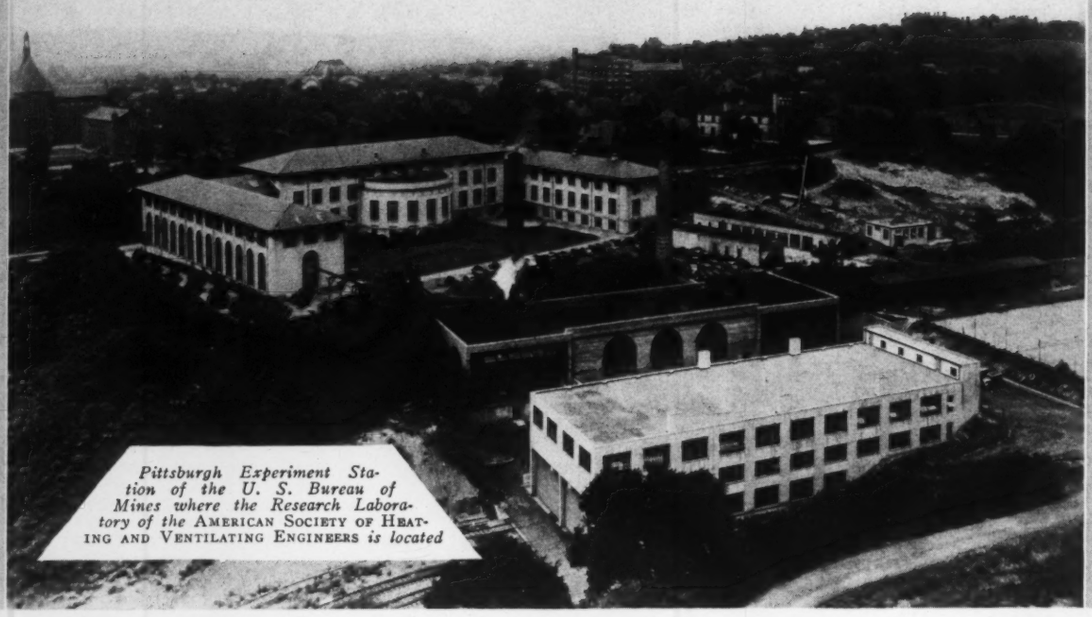
Plans for the Society's Semi-Annual Meeting to be held at the New Ocean House, Swampscott, Mass., are actively under way, and Massachusetts Chapter will act as host.

The dates selected are June 24, 25, and 26, 1937 and a pleasant three days are assured by the Committee on Arrangements.

W. T. Jones is General Chairman and he is assisted by the following chairmen of sub-committees: E. A. Dusossoit—*Ladies;* T. F. McCoy—*Transportation;* J. F. Tuttle—*Golf;* James Holt—*Entertainment*; A. S. Kellogg—*Registration and Reception;* R. M. Nee—*Publicity;* and W. E. Barnes—*Finance.*

It is planned to have the Council and Committee meetings on June 23 and technical sessions on each of the three following days. A tentative program of entertainment has been planned and some novel events are in prospect. Further announcements will be made to members soon.

Mark the dates, June 24-26, on the calendar!



Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS *is located*

COMMITTEE ON RESEARCH

W. A. DANIELSON, *Chairman;* W. L. FLEISHER, *Vice-Chairman;* F. C. HOUGHTEN, *Director;* DR. A. C. WILLARD, *Technical Adviser;* O. P. HOOD, *Ex-Officio Member. One Year:* C. A. DUNHAM, W. L. FLEISHER, ELLIOTT HARRINGTON, A. P. KRATZ, H. C. MURPHY. *Two Years:* W. A. DANIELSON, C. E. LEWIS, D. W. NELSON, C. TASKER, C.-E. A. WINSLOW. *Three Years:* H. E. ADAMS, A. E. STACEY, G. L. TUVE, J. H. VAN ALSBURG, J. H. WALKER.

Executive Committee: W. A. Danielson, *Chairman;* W. L. Fleisher, J. H. Walker.

Technical Advisory Committees

Committee on Air Cleaning—IP-4: H. C. Murphy,* *Chairman;* M. I. Dorfan, C. E. Lewis,* S. R. Lewis, G. W. Penney, A. L. Simison, W. O. Vedder.

Committee on Air Conditioning Requirements of Glass—IF-18: M. L. Carr, *Chairman;* F. L. Bishop, A. N. Finn, E. H. Hobbie, R. J. Lillibridge, R. A. Miller, F. W. Parkinson, W. C. Randall, J. T. Staples, C. Tasker,* G. B. Watkins, F. C. Weinert.

Committee on Air Distribution—IP-21: Ernest Szekely, *Chairman;* S. H. Downs, M. K. Fahnestock, F. J. Kurth, D. W. Nelson,* C. H. Randolph, J. E. Schoen, G. L. Tuve,* J. H. Van Alsburg.*

Committee on Air Friction—IP-6: J. H. Van Alsburg,* *Chairman;* C. A. Booth, S. H. Downs, C. M. Humphreys, R. D. Madison, L. B. Miller, L. G. Miller.

Committee on Atmospheric Impurities and Resulting Safety and Health Requirements—IP-26: Theodore Hatch, *Chairman;* J. J. Bloomfield, C. A. Booth, Philip Drinker, Dr. Leonard Greenburg, Elliott Harrington,* H. B. Meller.

Committee on Climate and Air Conditioning—C-24: Dr. C. A. Mills, *Chairman;* Major G. C. Dunham, James Govan, Dr. W. J. McConnell, C. F. Neergaard, Dr. F. M. Pottenger, Jr., E. L. Weber, Prof. C.-E. A. Winslow.*

Committee on Comfort Air Conditioning—OH-22: C. Tasker,* *Chairman;* A. E. Beals, F. R. Bichowsky, Thomas Chester, F. E. Giesecke, Elliott Harrington,* Dr. E. V. Hill, R. E. Keyes, C. P. Yaglou.

Committee on Corrosion in Air Conditioning Equipment—IF-14: A. E. Stacey, Jr.,* *Chairman;* A. F. L. Anderson, M. L. Diver, F. L. LaQue, C. E. Lewis,* R. M. Palmer, F. N. Speller, C. M. Sterne, R. T. Thornton, J. H. Young.

Committee on Corrosion in Steam Systems—IF-2: J. H. Walker,* *Chairman;* E. L. Chappell, W. H. Driscoll, C. A. Dunham,* L. B. Miller, R. R. Seeber, C. M. Sterne.

Committee on General Air Conditioning Comfort Requirements—OH-5: C. P. Yaglou, *Chairman;* J. J. Aeberly, R. R. Sayers, C.-E. A. Winslow.*

Committee on Heat Requirements of Buildings—IF-8: O. W Armspach, *Chairman;* P. D. Close, W. H. Driscoll, H. M. Hart, P. E. Holcombe, V. W. Hunter, H. H. Mather, E. C. Rack, F. B. Rowley, R. J. J. Tennant, J. H. Walker.*

Committee on Heat Transfer of Finned Tubes with Forced Air Circulation—IP-10: F. B. Rowley, *Chairman;* H. F. Hutzel, C. E. Lewis,* E. J. Lindseth, R. H. Norris, W. E. Stark, G. L. Tuve.*

Committee on Insulation—IF-23: L. A. Harding, *Chairman;* E. A. Allcut, J. D. Edwards, E. C. Lloyd, R. T. Miller, E. R. Queer, T. S. Rogers, F. B. Rowley, W. S. Steele, C. Tasker,* B. Townshend, G. B. Wilkes.

Committee on Intermittent Heating—IP-20: E. K. Campbell, *Chairman;* W. L. Cassell, Prof. E. F. Dawson, N. W. Downes, F. E. Giesecke, J. M. Robertson, J. H. Kitchen, Prof. A. H. Sluss, G. L. Tuve.*

Committee on Psychrometry—C-11: F. R. Bichowsky, *Chairman;* C. A. Bulkeley, J. A. Goff, Dr. E. V. Hill, F. G. Keyes, A. P. Kratz,* W. M. Sawdon.

Committee on Radiation with Gravity Air Circulation—IP-9: M. K. Fahnestock, *Chairman;* H. F. Hutzel, J. P. Magos, J. W. McElgin, J. F. McIntire, D. W. Nelson,* R. N. Trane, T. A. Novotney.

Committee on Relation of Body Changes to Air Changes—OH-3: Dr. E. V. Hill, *Chairman;* N. D. Adams, J. J. Aeberly, John Howatt, A. P. Kratz,* P. J. Marschall, V. L. Sherman.

Committee on Sound Control—IF-1: J. S. Parkinson, *Chairman;* C. M. Ashley, V. O. Knudsen, R. F. Norris, C. H. Randolph, J. P. Reis, A. E. Stacey,* F. R. Watson.

Committee on Summer Air Conditioning for Residences—IP-7: M. K. Fahnestock, *Chairman;* E. A. Brandt, John Everetts, Jr., Elliott Harrington,* H. F. Hutzel, E. D. Milener, K. W. Miller, E. B. Newill, F. G. Sedgwick, J. H. Walker.*

Committee on Treatment of Air with Electricity—C-17: Prof. C.-E. A. Winslow,* *Chairman;* R. D. Bennett, W. H. Carrier, L. W. Chubb, Major W. D. Fleming, R. F. James, L. R. Koller, Dr. C. A. Mills, Prof. E. B. Phelps, Prof. G. R. Wait, Prof. W. T. Wells.

*Member of Committee on Research.

A. S. H. V. E. RESEARCH

A Page of News Notes and Comment

By **Brewster S. Beach***

LABORATORY STAFF BEING ENLARGED

As the year enters its second quarter, the A.S.H.V.E. Research Laboratory is prepared to extend its activities with an augmented staff of active workers as shown in the accompanying photograph taken on the steps of the Bureau of Mines on March 10.

Encouraged by the upturn from depression conditions, F. C. Houghten, director of the laboratory, looks forward to an unusually active period of research based on the program mapped out by the Committee on Research.

DISTRICT HEATING ASSOCIATION EXHIBIT

J. H. Walker

On recommendation of J. H. Walker, Detroit, member of the Committee on Research and chairman of the Technical Advisory Committee on Corrosion in Heating Systems, the Society's laboratory will have an exhibit at the annual meeting of the *National District Heating Association* to be held in Detroit, May 25-28.

One of the features of the laboratory exhibit is expected to be the Nicholls Heat Flow Meter, an instrument placed against wall surfaces to indicate instantaneous heat flow.

There will also be an exhibit showing relative heat transmission from steel and copper pipe. This consists of an electric boiler supplying steam to three 10 ft vertical pipes, one a 1 in. bare copper pipe, the second a 1 in. bare black iron pipe and the third a 1 in. iron pipe covered with insulation. The relative heat emission from the three pipes may be shown by the rate of condensate drip into graduated glass cylinders.

Plans are also being discussed for displaying the laboratory's special balance to measure loss of weight due to metabolism or exhalation of carbon dioxide resulting from oxidation of food and evaporation of moisture from the body surface and respiratory tract. This instrument used by the A.S.H.V.E. Laboratory, is of particular interest inasmuch as it is sensitive enough to indicate change in weight of persons to an accuracy of approximately one-fiftieth of a gram.

*Director, Information Service, Committee on Research.

GLASS COMMITTEE BEGINS ACTIVE OPERATIONS

M. L. Carr

Preliminary investigations of glass building units of interest to the Technical Advisory Committee on Air Conditioning Requirements of Glass are going forward at the Pittsburgh Testing Laboratories under the direction of M. L. Carr, committee chairman.

Data uncovered by these investigations, and others embracing various types of windows, will enable the Committee on Research to review, check, correlate and make available more complete technical data for the use of the engineer concerned with the use of glass in building. Importance of the A.S.H.V.E. research on glass results from the greater emphasis placed on controlled indoor air conditions following in the wake of the tremendous growth of air conditioning.

Complete information concerning the effect of the window on the inside environmental conditions is considered one of the most vital problems before the Committee on Research. The modern trend in air conditioning points to new developments in one of two directions—either the elimination of the window entirely with complete dependence on artificial illumination, or the construction of better windows including double glazing and in some cases triple glazing.

The Research Laboratory and the technical committee in charge of this work has a new outlook on the possibilities for handling so complicated a problem and is perfecting plans to pool all of the available data into a comprehensive report. First steps in this direction were taken in Pittsburgh last month.

WHEN IS A DRAFT NOT A DRAFT?

Or rather where is a draft a draft? The young lady in this picture is helping the A.S.H.V.E. Laboratory at Pittsburgh determine the physiological effect of a current of air at a determined temperature and velocity directed at the ankle.

This is a phase of the work of the Technical Advisory Committee on Comfort Air Conditioning of which C. Tasker, of Toronto, is chairman.

Governor of Rome reading his speech of welcome, International Congress for Technical Education

International Congress for Technical Education

For the International Congress for Technical Education held in Rome, Italy, December 28, 29 and 30, 1936, the A.S.H.V.E. appointed two delegates, Aldo Gini and C. F. Hauss of Milan. Mr. Gini was prevented from attending by illness, but Mr. Hauss was present and has forwarded an outline of the Congress together with the accompanying photographs.

The delegates first assembled in the large hall in the Campidoglio, an old but beautiful building which was once the Senate of ancient Rome, but now serves as the City Hall. Following a formal welcome given by the Governor of Rome in the name of the Duce, wreaths were laid paying tribute to the unknown soldier and to those who fell in the Fascist revolution. Motoring to the other side of Rome to the University City, the representatives were greeted by the Minister of Public Instruction.

Of the 1,500 delegates, approximately 1,000 were from Italy and the remainder came from twenty-six countries. The United States was officially represented by the Commercial Attaché of the American Embassy.

The program included consideration of Terminology and Special Forms for Medical Examinations, which were held over from the 1934 meeting in Barcelona. The matters then discussed were as follows: First Problem—(a) Technical education in the economic life of the state, (b) Technical education of juvenile unemployed; Second Problem—(a) Vocational guidance and its follow up, (b) Manual guidance as an indication of aptitude, (c) Is some change necessary in preparing youth during their time of growth?, (d) The attitude toward specific vocations; Third Problem—Special training for workshop instructors charged with the duty of practical instruction in vocational, technical schools and colleges; Fourth Problem—Training of women for their special place in economic life, (a) In the home, (b) Elsewhere; Fifth Problem—Miscellaneous matters.

International Congress in session

President Boyden and Vice-President Gurney to Visit Chapters

During April and May arrangements have been made for the official visits of Pres. D. S. Boyden and 1st Vice-Pres. E. H. Gurney to local Chapters of the Society.

At Washington, D. C., April 9, a meeting of the Council will be held at Hotel Harrington, after which all the Officers and Council members will attend the Washington, D. C. Chapter meeting and the featured speakers of the evening will be Pres. Davis S. Boyden, Boston, 1st Vice-Pres. E. Holt Gurney, Toronto, and W. A. Danielson, Washington, D. C., chairman of the Committee on Research and president of the Washington, D. C. Chapter.

President Boyden will then visit St. Louis, April 12th; Kansas City, April 13th; Oklahoma City, April 14th; Dallas, Tex., April 15th; Los Angeles, April 20th; Seattle, Wash., April 30th; Winnipeg, Man., May 5th; Minneapolis, May 7th; Chicago, May 10th, and Milwaukee, May 11th.

During April Mr. Gurney will visit Pittsburgh on the 19th, Cincinnati on the 20th, Cleveland on the 21st, Detroit on the 22nd, and in May he will be in Buffalo on the 17th.

Resolution of Appreciation to F. C. Houghten

The following resolution was adopted by the Council at its meeting on January 24, 1937, and presentation was made by Pres. D. S. Boyden during the 43rd Annual Meeting of the Society in St. Louis. The resolution was prepared by a special committee appointed by the Council, consisting of A. P. Kratz, *chairman*, C. V. Haynes and John Howatt.

Whereas,

Ferry C. Houghten

has served as

Director of the Research Laboratory

since September 1925, and throughout this time, which includes the period when the problems were most trying because of the shrinkage of funds available for research, has conscientiously devoted himself to carrying the burdens and discharging the responsibilities of that Directorship; and in addition has cheerfully served the Society in many ways by assuming duties not directly entailed in the discharge of the responsibilities of his office; and

Whereas,

He has directed the research of

The American Society of Heating and Ventilating Engineers

so that it has been recognized as a most important activity of the Society and the greatest contribution to the development and progress in the arts and science of Heating, Ventilating and Air Conditioning, and has contributed many papers which have been of great scientific value to the engineering profession; therefore be it

Resolved,

That the Council at this time express its appreciation of the services rendered by Ferry C. Houghten during the years that he has been Director of the Research Laboratory, and that a copy of these resolutions be presented to him and spread upon the records of the Council.

A. V. Hutchinson, Secretary — G. L. Larson, President

January 24, 1937

NEWS OF LOCAL CHAPTERS

OFFICERS OF LOCAL CHAPTERS—1937

CINCINNATI: *Organized,* 1932. *Headquarters,* Cincinnati, O. *Meets,* Second Tuesday in Month. *President,* E. B. ROYER, 6635 Iris Ave. *Secretary,* O. W. MOTZ, 920 E. McMillan St.

CLEVELAND: *Organized,* 1916. *Headquarters,* Cleveland, O. *Meets,* Second Thursday in Month. *President,* L. T. AVERY, 2341 Carnegie Ave. *Secretary,* W. R. BEACH, 75 Public Sq.

ILLINOIS: *Organized* 1906. *Headquarters,* Chicago, Ill. *Meets,* Second Monday in Month. *President,* J. J. HAYES, 53 W. Jackson Blvd. *Secretary,* C. E. PRICE, 6 N. Michigan Ave.

KANSAS CITY: *Organized,* 1917. *Headquarters,* Kansas City, Mo. *Meets,* Second Monday in Month. *President,* L. R. CHASE, 2440 Pennway. *Secretary,* F. J. DEAN, JR., 6028 Walnut St.

MANITOBA: *Organized,* 1935. *Headquarters,* Winnipeg, Man. *Meets,* Fourth Thursday in Month. *President,* J. B. STEELE, 184 Waterloo St. *Secretary,* C. H. TURLAND, 325 Centennial St.

MASSACHUSETTS: *Organized,* 1912. *Headquarters,* Boston, Mass. *Meets,* Second Tuesday in Month. *President,* JAMES HOLT, Massachusetts Institute of Technology, Cambridge, Mass. *Secretary,* JOHN TURNER, 285 Columbus Ave.

MICHIGAN: *Organized,* 1916. *Headquarters,* Detroit, Mich. *Meets,* First Monday after the 10th of the Month. *President,* R. K. MILWARD, 127 Campbell Ave. *Secretary,* G. H. TUTTLE, 2000 Second Ave.

WESTERN MICHIGAN: *Organized,* 1931. *Headquarters,* Grand Rapids, Mich. *Meets,* Second Monday in Month. *President,* L. G. MILLER, Mich. State College, E. Lansing, Mich. *Secretary,* E. I. ADAMS, 115 So. Pine, Lansing, Mich.

MINNESOTA: *Organized,* 1918. *Headquarters,* Minneapolis, Minn. *Meets,* Second Monday in Month. *President,* N. D. ADAMS, 220 Second Ave. S. W., Rochester, Minn. *Secretary,* J. E. SWENSON, 800 Hennepin Ave.

MONTREAL: *Organized,* 1936. *Headquarters,* Montreal, Que. *Meets,* Third Monday in Month. *President,* G. L. WIGGS, University Tower. *Secretary,* C. W. JOHNSON, 630 Dorchester St., W.

NEW YORK: *Organized,* 1911. *Headquarters,* New York, N. Y. *Meets,* Third Monday in Month. *President,* G. E. OLSEN, 101 Park Ave. *Secretary,* T. W. REYNOLDS, 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

WESTERN NEW YORK: *Organized,* 1919. *Headquarters,* Buffalo, N. Y. *Meets,* Second Monday in Month. *President,* P. S. HEDLEY, Curtiss Bldg. *Secretary,* C. A. GIFFORD, 758 Parkside Ave.

OKLAHOMA CITY: *Organized,* 1935. *Headquarters,* Oklahoma City, Okla. *Meets,* Second Monday in Month. *President,* F. X. LOEFFLER, 1604 N. W. Fifth St. *Secretary,* E. W. GRAY, Box 1498.

ONTARIO: *Organized,* 1922. *Headquarters,* Toronto, Ont. *Meets,* First Monday in Month. *President,* THOMAS MCDONALD, 117 Peter St. *Secretary,* H. R. ROTH, 57 Bloor St. W.

PACIFIC NORTHWEST: *Organized,* 1928. *Headquarters,* Seattle, Wash. *Meets,* Second Tuesday in Month. *President,* S. D. PETERSON, 473 Colman Bldg. *Secretary,* D. C. GRIFFIN, Orpheum Bldg.

PHILADELPHIA: *Organized,* 1916. *Headquarters,* Philadelphia, Pa. *Meets,* Second Thursday in Month. *President,* W. F. SMITH, 422 Bryn Mawr Ave., Cynwyd, Pa. *Secretary,* H. H. ERICKSON, 804 Architects Bldg.

PITTSBURGH: *Organized,* 1919. *Headquarters,* Pittsburgh, Pa. *Meets,* Second Monday in Month. *President,* M. L. CARR, P. O. Box 1646. *Secretary,* T. F. ROCKWELL, Carnegie Inst. Tech.

ST. LOUIS: *Organized,* 1918. *Headquarters,* St. Louis, Mo. *Meets,* First Tuesday in Month. *President,* C. R. DAVIS, 2328 Locust St. *Secretary,* R. J. TENKONOHY, 1474 S. Vandeventer.

SOUTHERN CALIFORNIA: *Organized,* 1930. *Headquarters,* Los Angeles, Calif. *Meets,* Second Tuesday in Month. *President,* LEO HUNGERFORD, 1057 N. La Brea Ave., Hollywood, Calif. *Secretary,* H. M. HENDRICKSON, 5051 Santa Fe Ave.

TEXAS: *Organized,* 1936. *Headquarters,* College Station, Texas. *President,* F. E. GIESECKE, Texas A. & M. College. *Secretary,* W. H. BADGETT, Texas A. & M. College.

WASHINGTON, D. C.: *Organized,* 1935. *Headquarters,* Washington, D. C. *Meets,* Second Wednesday in Month. *President,* W. A. DANIELSON, 2236 Munitions Bldg. *Secretary,* M. D. KICZALES, 3000 Connecticut Ave., N. W.

WISCONSIN: *Organized,* 1922. *Headquarters,* Milwaukee, Wis. *Meets,* Third Monday in Month. *President,* M. E. ERICKSON, 8405 W. Greenfield Ave., West Allis, Wis. *Secretary,* R. G. KOCH, 626 E. Wisconsin Ave.

Montreal Chapter Receives Charter

February 15, 1937. The regular meeting of the Montreal Chapter was held at the Windsor Hotel with 39 members and guests attending. Following dinner, the meeting was called to order by Pres. G. L. Wiggs and the guests were presented.

The minutes of the previous meeting were read by Secy. C. W. Johnson and approved.

President Wiggs introduced E. H. Gurney, Toronto, 1st vice-president of the A.S.H.V.E., who made the presentation of the Charter to the Montreal Chapter on behalf of the officers and members of the Council. Mr. Gurney gave a short review of A.S.H.V.E. membership and Society activities.

L. H. Laffoley and President Wiggs reported on the Society's annual meeting in St. Louis which they attended.

Mr. Gurney was the principal speaker of the evening and gave a very interesting talk on Heating and Air Conditioning Observations in South Africa.

W. L. Fleisher, New York, N. Y., who was a guest of the Chapter, was called on by President Wiggs to give a general talk on the papers presented at the St. Louis Annual Meeting.

A vote of thanks was extended on behalf of the Chapter to the speakers by W. W. Timmins.

An attendance card system was inaugurated at this meeting and F. G. Phipps, treasurer, requested immediate payment of local dues.

Southern California Chapter Holds Joint Meeting with A. S. R. E.

February 16, 1937. The A. S. H. V. E. Southern California Chapter held a joint meeting with local members of the *A. S. R. E.* at the Rosslyn Hotel. Dinner was served at 6:15 p. m. with 77 members and guests present.

Following the dinner, Chairman J. C. Blair of *A. S. R. E.* called the meeting to order and welcomed the members of both Societies and guests. The meeting was then turned over to Pres. Leo Hungerford of A. S. H. V. E., who expressed the pleasure of the Chapter in the joint meeting and announced similar plans for March.

President Hungerford called on Secy. H. M. Hendrickson to report on the 43rd Annual Meeting of the A. S. H. V. E. which he attended as representative of Southern California Chapter. Secretary Hendrickson also read a letter from the Department of Commerce concerning the availability of a government report on plumbing and heating equipment markets and a communication regarding employment.

O. W. Ott was introduced and spoke in regard to the Society's Annual Meeting and the local research project. Mr. Ott expressed the hope that research work could soon be started at the University of California.

E. H. Kendall, chairman of the Program Committee, announced that Pres. D. S. Boyden of the A. S. H. V. E. was planning to visit Southern California Chapter in the early Spring.

The meeting was then turned back to Chairman Blair and an entertaining musical program was presented through the courtesy of the Pacific Scientific Co. W. L. Holladay was called upon to tell about the program for next month's meeting at the Clark Hotel as a joint meeting with the Founder Societies in honor of Secy. D. L. Fiske's visit to the local *A. S. R. E.* Section.

P. H. Thompson, chief engineer, Alco Valve Co., St. Louis, Mo., was the speaker of the evening and was introduced by Mr. Fowler. Mr. Thompson gave an interesting talk, illustrated with slides, on the subject of Refrigerant Control Valves, and pro-

vided an instructive lecture on the construction of expansion valves and the application problems encountered in the field.

Following Mr. Thompson's talk the meeting adjourned at 10:00 p. m.

January 12, 1937. Thirty-seven members and guests were present at a luncheon meeting of Southern California Chapter at the Rosslyn Hotel, at which Secy. H. M. Hendrickson presided.

A telegram from Pres. C. R. Davis of St. Louis Chapter was read, inviting Southern California Chapter members to the Society's Annual Meeting in St. Louis.

Treas. A. W. Cooper was called upon for a financial report and O. W. Ott, chairman of the Research Committee, reported that further progress relative to the proposed research work at Berkeley pended action of the Society's Council.

E. H. Kendall, chairman of the Program Committee, was called upon to introduce the speaker, J. K. Horton, a Los Angeles attorney. The subject of Mr. Horton's interesting speech was The Robinson-Patman Act and Its Relationship to the Contracting and Sub-contracting Industries. In the discussion which followed, the members asked Mr. Horton many questions regarding various phases of the Act.

The meeting then adjourned at 1:30 p. m.

Glass Blocks Provide Interesting Subject for Washington Members

February 10, 1937. Washington, D. C. Chapter held its regular meeting at the Hotel Harrington with 35 members and guests present at dinner and 60 at the meeting, which followed.

The business session was called to order by Pres. W. A. Danielson at 8:00 p.m., and the minutes of the preceding meeting of the Chapter and of the Board of Governors were read and approved.

E. H. Lloyd, chairman of the Membership Committee, reported that the Committee is planning an aggressive campaign for new members.

On motion duly made and seconded, it was resolved that the Chapter go on record as favoring a joint meeting with the local chapter of the *A.S.M.E.* at a date to be decided by the Board of Governors.

President Danielson reported on the papers presented at the Annual Meeting of the A.S.H.V.E. in St. Louis and then discussed briefly his work as chairman of the Society's Committee on Research.

President Danielson then introduced R. A. Miller, of Pittsburgh Plate Glass Co., who gave an interesting talk, The Insulation Value and the Light Transmission Value of Glass Blocks. Mr. Miller prefaced his remarks with a history of the manufacture of glass, dating back to the Egyptian era. The world's glass manufacturing center was at one time in Rome, and after the burning of Rome moved to Venice. In the seventeenth century England became the world's center for the manufacture of glass, and later France. The methods of manufacture were being constantly improved and Mr. Miller showed, with the use of lantern slides, how plate glass is now manufactured in this country. With modern American methods, the speaker stated that this country now manufactures the finest glass products in the world.

Mr. Miller then conducted an open forum on the subject and many questions were asked about glass blocks for building construction, their heat transmission and insulating value.

There being no further business to come before the meeting, Secy. M. D. Kiczales reports that the session adjourned at 10:10 p. m.

Air Conditioning with Ice Subject of St. Louis Meeting

February 16, 1937. First Vice-Pres. G. W. F. Myers called the meeting of St. Louis Chapter to order at 7:35 p. m. at Garavelli's Restaurant, with 15 members and 10 guests present. The minutes of the previous meeting were read and approved.

Secy. R. J. Tenkonohy brought up the matter of the invoice received from the Joint Council of the *Associated Engineering Societies of St. Louis,* which was approved for payment. Secretary Tenkonohy also read the announcement of the Cleveland Chapter meeting, February 12, and letters from headquarters office in regard to membership.

The treasurer, C. E. Hartwein, reported a balance on hand of $355.78.

D. J. Fagin reported for the Program Committee and stated that a joint meeting had been arranged for March with the Engineers Club and the local section of the *A.S.R.E.,* with Prof. A. P. Kratz, University of Illinois, as speaker.

Vice-President Myers of the Membership Committee announced that 10 chapter members had affiliated since January 1, 1936, that 3 more are expected shortly, and that 6 A.S.H.V.E. membership applications are before the Committee on Admission and Advancement.

Vice-President Myers called on all chairmen of the Committee on Arrangements for the A.S.H.V.E. Annual Meeting who gave oral reports on the activities of their committees, and on Secretary Tenkonohy for a written report of the General Arrangements Committee.

After a brief recess the meeting was resumed at 8:25 p. m. with 30 attending when Mr. Fagin presented C. F. Boester, who spoke on Air Conditioning With Ice.

Mr. Boester passed to each in the audience three blueprints illustrating different methods of application, and discussed their merits as they affected performance, operation and first costs. He stated that a 10-ton load was the dividing line for overall economy; that smaller plants were not practical; and that installations in northern latitudes were more frequent. Mr. Boester said that in conditioning air with ice first cost is more favorable than with other systems, and that it is found desirable to sell effective temperature rather than design temperature. After a spirited discussion, with numerous questions ably answered, Mr. Boester closed his talk at 9:10 p. m.

Mr. Fagin returned the Chair to Vice-President Myers, who extended to Mr. Boester a rising vote of thanks of the audience, and adjourned the meeting.

Air Conditioning Symposium Continued at Pittsburgh

March 8, 1937. Pres. M. L. Carr called the Pittsburgh Chapter meeting to order at 7:45 p. m. in the Downtown Y. M. C. A. with approximately 60 members and guests present. The minutes of the February meeting were read and approved.

Treas. J. E. McLean presented a report on the financial condition of the Chapter and J. F. S. Collins, Jr., of the Program Committee, reported that the May and June meetings would be devoted to a symposium on Atmospheric Pollution.

Chairman P. A. Edwards of the Membership Committee stated that the applications of two candidates had been filed.

As chairman of the Current Events Committee, T. F. Rockwell introduced H. A. Krintzman, senior student at Carnegie Institute of Technology. Mr. Krintzman discussed briefly the *A.S.R.E.* proposed code for rating mechanical condensing units and then discussed a series of charts which showed under what conditions of condenser pressure and evaporator temperature it was possible for freon condensing units to have a capacity of a ton of refrigeration per horsepower input.

The meeting was then turned over to R. J. J. Tennant, who again presided at the Symposium on Air Conditioning an Office Building. Mr. Tennant introduced the following speakers: J. L. McCullough, who spoke on the selection of direct radiation to carry part of the heating load in severe weather; E. C. Smyers, who discussed the control of direct radiation by means of an outside thermostat; F. C. McIntosh, who completed his dis-

cussion of the control of the air conditioning system during the heating season; P. A. Edwards, who presented an estimate of the cost of the piping for the radiator system; V. W. Hunter, who discussed the operating, maintenance, and investment problems arising in connection with air conditioning large office buildings.

Mr. Tennant then closed the program by presenting a cost analysis of the complete air conditioning system, which indicated that the owner should receive an additional rental of 20c to 25c per square foot per month to take care of the additional investment and operating expenditures resulting from the adoption of air conditioning.

The meeting then adjourned at 10:45 p. m.

Cincinnati Chapter Discusses Automatic Controls as Applied to Air Conditioning

March 9, 1937. The regular meeting of Cincinnati Chapter was held at the Hotel Gibson with 30 members and guests present.

Pres. E. B. Royer spoke briefly regarding the A.S.H.V.E. Annual Meeting in St. Louis and reported on the conference of chapter delegates. He urged everybody who could to attend the Annual and Semi-Annual Meetings of the Society.

President Royer stated that the invitation of the Cincinnati Chapter to entertain the A.S.H.V.E. at its Annual Meeting in January, 1939, had been presented and was being given consideration by the Council.

G. B. Houliston, chairman of the Membership Committee, said that he had a new supply of application blanks, and urged that the effort to secure new members be maintained.

A. B. Newton, control systems development engineer, Minneapolis-Honeywell Regulator Co., was introduced and gave an interesting and instructive talk, illustrated with lantern slides, on Automatic Control as Applied to Air Conditioning.

Mr. Newton gave an explanation of the Newton psychrometric chart, showing how various cooling cycles operate and require both thermostats and humidistats for control at light loads. Mr. Newton explained that consideration should be given not only to the maximum cooling load but also to light loads, if excessive humidity is to be avoided on moderately warm but humid days. He gave examples of systems requiring reheat in the rooms during such periods of light load, and in stores, where the lights or radiators have been used to maintain comfortable temperatures when the required dehumidification was secured.

The latter part of the presentation included pictures and descriptions of 15 standard control systems for air conditioning systems of the various types, embodying temperature and humidity control and operation of outside and return air dampers to give economic and satisfactory service under varying outdoor and indoor conditions.

President Royer complimented the speaker on the completeness of the exposition and the meeting adjourned with a vote of thanks to Mr. Newton.

Social Meeting for Cleveland Members

February 12, 1937. About 90 members and guests attended the meeting of Cleveland Chapter at the Germania Turnverein.

Pres. L. T. Avery called the meeting to order at 8:50 p. m. and dispensed with the customary routine business. F. A. Rodgers, chairman of the Program Committee, introduced the speaker, Jack Graney of WHK.

Mr. Graney gave an interesting play by play account of the 1920 World Series, during which Elmer Smith made a home run with the bases full, and Bill Wamby made a triple play unassisted. Both Mr. Wamby and Mr. Smith were present.

A short play was then given by four members, which was interesting and entertaining.

Refreshments were served and a pleasant evening was enjoyed by all, according to Secy. W. R. Beach.

Death of Prof. Elihu Thomson

Prof. Elihu Thomson, 83, dean of American scientists, and one of the founders of the General Electric Co., died at his home in Swampscott, Mass., March 13. Together with Thomas A. Edison, James J. Wood, and Charles F. Brush, Professor Thomson was one of the group who created the modern electrical industry.

Professor Thomson was one of America's greatest pioneers in the field of electrical science, and he held upward of 700 patents in the United States alone. He originated the resistance method of electric welding; developed the repulsion type of induction electric motor; invented the magnetic blow-out principle in lightning arresters and electric switches, the oil-cooled type of transformer, the constant-current transformer, and the modern process of commercially treating fused quartz. He was the originator of the three-phase electric dynamo machine in 1879 and he invented the first practical wattmeter, thereby winning half of the grand prize offered at the close of the Paris electrical exposition of 1889.

Professor Thomson had the distinction of being the only man ever to receive all three of England's highest scientific honors—the Hughes, the Lord Kelvin, and the Faraday medals. He also had the medal of the Verein Deutscher Ingenieure, outstanding German engineering award, the John Fritz medal, and the medals of the Franklin Institute, the American Academy of Arts and Sciences and the *American Institute of Electrical Engineers.* He was president of the International Electrical Congress in 1904, and the president of the Electro-chemical Congress in 1910 and 1911. He also served as vice-president of the American Academy of Arts and Sciences and vice-president of the *American Philosophical Society.*

Alfred D. Flinn Dies

Dr. Alfred Douglas Flinn, director of the Engineering Foundation, New York, N. Y., died of heart disease at Heathcote Inn on March 14, 1937, after a long illness.

Dr. Flinn was born in New Berlin, Pa., August 4, 1869, and after graduation from the Worcester, Mass., School, he studied at the Massachusetts Institute of Technology. Before going to New York City he worked in the city engineer's office of Providence and with the Massachusetts Metropolitan Water Works.

From 1905-1918 Dr. Flinn was a member of the Board of Water Supply of the City of New York and he served successively as departmental engineer, deputy chief engineer and acting chief engineer. He helped to develop the city's Esopus and Schoharie water sheds and their reservoirs, dams and aqueducts. He was credited with the final design of aerators for the entire flow in the Catskill Aqueduct, which is a strategic part of the New York water system.

In 1918 Dr. Flinn became secretary of the United Engineering Society, which later was succeeded by the Engineering Foundation and United Engineering Trustees. He resigned as secretary of the latter organization in 1934. In his work with the Engineering Foundation, which he served as director since 1922, Dr. Flinn helped supervise and coordinate the research work of the related organizations, and, with the aid of the National Research Council, he established the Highway Research Board, American Bureau of Welding and the Personnel Research Federation.

From 1902 to 1904 Dr. Flinn was managing editor of *The Engineering Record,* which was combined later with *The Engineering News.* He compiled the *Waterworks Handbook* with R. S. Weston and C. L. Bogert and wrote many technical papers in addition to editing the *Research Narratives* of the Engineering Foundation.

Dr. Flinn had received many honors for his engineering work, among them being the honorary degree of Doctor of Applied Sciences from the University of Louvain, France, in 1927, honor-

ary Doctor of Engineering from Worcester (Mass.) Polytechnic Institute in 1932, and honorary member of the Masaryk Academy of Prague, as well as other Czechoslovakian decorations.

Dr. Flinn was director of the *American Society of Civil Engineers* from 1917 to 1918, and was past president of the *Municipal Engineers of the City of New York,* past secretary of the *Engineering Council* and past president of the *Personnel Research Federation.* He was assistant secretary of the John Fritz Medal Fund Corp. from 1920 to 1935 and was formerly secretary and treasurer of the Daniel Guggenheim Medal Fund. He was a fellow of the *American Association for the Advancement of Science,* and a member of the *American Mathematical Society, American Institute of Mining and Metallurgical Engineers, American Iron and Steel Institute, Engineers' Club, Century Association,* and Sigma Xi and Sigma Alpha Epsilon fraternities.

Surviving are his wife, Mrs. Mary Brownell Davis Flinn, a brother, Prof. F. B. Flinn, New York, N. Y., and a sister, Mrs. Arthur Ramsdall, Larchmont, N. Y.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the JOURNAL of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 61 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn the Council, urge the members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by April 15, 1937, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
ALLEN, C. V., Engrg. Mgr. Midwest Air Cond. Corp., St. Louis, Mo.	D. W. McLenegan Paul Sodemann	Elliott Harrington S. R. Lewis
BADARACCO, J. A., Owner, Badaracco Appliance Co., Mexico, Mo.	C. R. Davis G. W. F. Myers	A. L. Haller R. J. Tenkonohy
BENSON, B. C., Br. Mgr. Sales Engrg., American Radiator Co., Chicago, Ill.	H. G. Kreissl C. W. De Land	R. E. Hattis John Howatt
BIBER, H. A., Asst. Operating Engr., Mellon National Bank, Pittsburgh, Pa.	P. A. Edwards G. G. Waters	H. L. Moore R. B. Stanger
BLACK, F. M., Chief Engr., Army Medical Center, Washington, D. C.	Glegge Thomas W. A. Danielson	C. R. Seckinger (*Non-Member*) M. S. Dickson (*Non-Member*)
BLACKSHAW, J. L., Engr., Air and Refrigeration Corp., New York, N. Y. (*Advancement*)	W. L. Fleisher John Everetts, Jr.	F. C. Houghten G. L. Wiggs
BODMER, EMMANUEL, Oil Heat Engr., Ets. Dieny et Lucas, Paris, France	Earl Nesmith A. T. Simmons (*Non-Member*)	R. D. Marshall (*Non-Member*) O. H. Price (*Non-Member*)
BORAK, EUGENE, Chief Draftsman, Buensod-Stacey Air Cond., Inc., New York, N. Y.	A. C. Buensod A. E. Stacey, Jr.	M. S. Smith H. L. Janet
BRAYMAN, A. I., Htg., Edward Brayman, Boston, Mass.	M. G. Knowles E. A. King (*Non-Member*)	C. W. Banks (*A. S. C. E.*) L. F. Carton (*Non-Member*)
BROWN, J. S., Jr., Sales Engr., Delco Frigidaire Cond. Div., Dayton, Ohio.	C. E. Lewis F. H. Williams	J. J. La Salvia A. R. MacMillan
BURR, G. C., Sales Engr., Dail Steel Prod. Co., Hyde Park, N. Y.	B. F. McLouth M. S. Hall	E. B. Root L. G. Miller
CARBONE, J. H., Htg. & Vtg. Engr., L. J. Wing Mfg. Co., New York, N. Y.	H. S. Wheller A. E. Seelig	C. L. Meyer J. H. Vogt
CARLSON, C. V., Jr. Engr., Sidles Co., Airtemp Div., Lincoln, Nebr.	M. J. Stevenson F. E. Prawl	Arledge Stokes L. L. Daubert
COOKE, T. C., Sales, Tomlinson Co., Inc., Durham, N. C.	C. A. Rueger, Jr. G. F. Hackney (*Non-Member*)	J. H. Verell (*Non-Member*) Hunter Jones (*Non-Member*)
CRUMP, A. L., Sales Engr., Powers Regulator Co., Chicago, Ill.	H. C. Mueller C. S. Leopold	M. W. Bishop C. E. Price
CUTHBERTSON, M. W., Supt. of Bldg., Hardware Mutual Fire Ins. Co., Minneapolis, Minn.	J. R. Roberts H. S. Morton	H. E. Gerrish D. L. Sutherland
DAMM, D. A., Air Cond. Engr., Gerken Heating Corp., Toledo, Ohio.	J. E. Maynard J. A. Schurman	R. W. Geltz J. Q. Treadway
DEAN, DAVID, Sales & Htg. Engr., New York Plumbers Spec. Co., New York, N. Y.	R. W. Cumming E. J. Ritchie	H. L. Hosking Jacob Miller
DE VILBISS, Air Cond. Engr., Griffith Amusement Co., Oklahoma City, Okla.	J. R. Hall, Jr. L. P. Hanson	B. P. Edelman A. E. Bauer
EISELE, W. S., Owner, Ideal Heating, Buffalo, N. Y.	W. E. Voisinet J. J. Yager	A. R. Kamman B. C. Candee
EWENS, F. G., Demonstrator in Thermodynamics, University of Toronto, Toronto, Ont., Can.	E. A. Allcut M. W. Shears	J. H. Fox Thomas McDonald

CANDIDATES

FOSTER, P. H., Mgr., Hudson Bay Plbg. Co., Flin Flon, Man., Can.

FRANCIS, P. E., Asst. Sales Mgr., North Western Fuel Co., Minneapolis, Minn.

FRIEDLINE, J. M., Sales Engr., Sidles Co., Airtemp Div., Des Moines, Iowa.

GERHARD, D. H., Power Sales Engr., Consumers Power Co., Jackson, Mich.

GREENLEAF, R. P., Air Cond. Engr., City Ice & Fuel Co., Cleveland, Ohio.

GUMAER, P. W., Consulting Engr., New York, N. Y.

HALL, J. R., Chief Engr., U. S. Air Cond. Corp., Minneapolis, Minn. (*Advancement*)

HECHLER, SAMUEL, Asst. Mgr., Westchester Square Plbg. Supply Co., New York, N. Y.

HEDLUND, R. A., Student, Purdue University, W. Lafayette, Ind.

HELLMERS, C. C., Jr., Sales Engr., Airtemp Div., Sidles Co., Lincoln, Nebr.

HESSELSCHWERDT, A. L., Jr., Instructor Mech. Engrg., Wayne University, Detroit, Mich.

IRWIN, R. R., Air Cond. Engr., York Ice Mach. Corp., St. Louis, Mo.

JEHLE, FERDINAND, Director of Research Lab., Hoffman Specialty Co., Stamford, Conn.

KILLIAN, V. J., Htg. Contractor, V. J. Killian Co., Winnetka, Ill.

KIMMEL, W. G., Sales Engr., York Ice Mach. Corp., Omaha, Nebr.

LANDEWIT, C. J., Combustion Engr., Combustion Maintainance Co., New York, N. Y.

LEE, J. A., Mgr. Refrigeration Div., Interstate Electric Co., Shreveport, La.

LEUPOLD, G. L., Sales Engr., Minneapolis-Honeywell Reg. Co., Cincinnati, Ohio.

LINDSAY, G. W., Jr., Design Engr., Frigidaire Div., Chicago, Ill.

LOUCKS, D. W., Supervisor, Com. Elec. & Steam Sales, Duquesne Light Co., Pittsburgh, Pa. (*Reinstatement*)

MANNING, C. E., Air Cond. Sales Engr., von Hamm-Young Co., Ltd., Honolulu, Hawaii.

MOORE, B. J., Jr., Archt., Griffith Amusement Co., Oklahoma City, Okla.

MULLOY, Edward, Mech. Supt., Brooklyn, N. Y.

OTTO, W. F., Chief Engr., Carson Pirie Scott & Co., Chicago, Ill.

PAVEY, C. A., Dist. Mgr., B. F. Sturtevant Co., Detroit, Mich.

PICKETT, C. A., Sales Repr., Herman Nelson Corp., Moline, Ill. (*Reinstatement*)

PLUMMER, R. S., Asst. Supt., Franklin Htg. Station, Rochester, Minn.

RYERSON, H. E., Mgr. Air Cond. Div., Bryant Air Cond. Corp., Chicago, Ill.

SCHMIDT, H. I., Experimental Engr., Fedders Mfg. Co., Inc., Buffalo, N. Y.

SCHULER, W. B., Direct Factory Repr., Taco Heaters, Inc., Chicago, Ill.

SHAFFER, C. E., Research Engr., Koppers Co., Pittsburgh, Pa.

SNAVELY, E. R., Chief Engr., Thermal Engrg. Co., Roselle, N. J.

SPENCER, E. D., Mgr. Commercial Dept., Brown Electric Co., Oklahoma City, Okla.

STEINKE, B. J., Draftsman, B. H. Steinke, New York, N. Y.

SUTFIN, G. V., Sales Engr., American Blower Corp., Cincinnati, Ohio.

WAGGONER, J. H., Research Engr., Owens-Illinois Glass Co., Muncie, Ind.

WAID, G. H., Sales Mgr., Scott Valve Mfg. Co., Detroit, Mich. (*Reinstatement*)

REFERENCES

Proposers	*Seconders*
D. F. Michie William Glass	R. L. Kent C. H. Turland
G. S. Bean W. F. Uhl	E. J. Uhl C. E. Gausman
Arledge Stokes M. J. Stevenson	F. E. Prawl L. L. Daubert
S. H. Downs R. W. Wilson	W. G. Schlichting C. R. McConner
W. R. Beach E. W. Gray	C. A. McKeeman H. M. Nobis
Leonard Greenburg R. R. Sayers (*Non-Member*)	W. P. Yant (*Non-Member*) John Roach (*Non-Member*)
L. P. Hanson A. E. Bauer	B. P. Edelman C. F. Hawkinson
H. J. Rose W. R. Zuhlke	O. O. Oaks F. B. Howell
J. D. Hoffman L. V. Ludy (*A. S. M. E.*)	G. A. Young (*A. S. M. E.*) W. T. Miller (*Non-Member*)
M. J. Stevenson F. E. Prawl	Arledge Stokes L. L. Daubert
James Holt J. R. Hertzler	W. E. Zieber C. L. Svenson (*Non-Member*)
G. W. F. Myers E. E. Carlson	L. J. DuBois H. W. Meinholtz
Potter Bowles C. V. Haynes	E. N. Sanbern C. W. Trambauer
J. R. Vernon J. H. Milliken	F. P. Keeney R. E. Hattis
L. J. Du Bois G. W. F. Myers	E. E. Carlson E. A. Grossman
Herbert Herkimer J. L. Blackshaw	John Everetts, Jr. W. L. Fleisher
H. F. Hutzel S. A. Whitt	W. G. Boales A. E. Knapp
K. W. Schick G. B. Houliston	O. W. Motz E. B. Royer
W. R. Knudsen J. R. Vernon	C. E. Price J. H. Milliken
J. F. S. Collins, Jr. W. W. Stevenson	R. J. J. Tennant F. A. Gunther
L. R. Chase M. A. Disney	E. K. Campbell Carl Clegg
J. R. Hall, Jr. L. P. Hanson	B. P. Edelman A. E. Bauer
John James John Everetts, Jr.	Benjamin Garfunkel (*Non-Member*) John G. Davis (*Non-Member*)
H. M. Hart J. J. Haines	M. W. Bishop J. H. Milliken
G. H. Cummins F. J. Feely	W. G. Boales E. H. Clark
C. R. Davis Paul Sodemann	C. M. Baumgardner S. S. Wilson (*Non-Member*)
H. D. Adams H. E. Gerrish	F. B. Rowley E. F. Jones
S. M. Trumbo Lloyd Howell	C. E. Price C. M. Burnam, Jr.
E. R. Walker Harry Schmidt	B. C. Candee J. Askin (*A. S. R. E.*)
C. M. Baumgardner J. R. Murphy	H. R. Linn E. M. Mittendorff
F. C. Houghten P. Nicholls	R. A. Sherman B. M. Bird (*A. I. M. M. E.*)
F. C. Houghten C. V. Haynes	H. W. Fiedler Clarke Redfield (*Non-Member*)
S. L. Rolland R. E. Pauling	E. W. Gray E. F. Dawson
W. R. Zuhlke M. C. Giannini	R. D. Lambert R. W. Cumming
D. L. Taze H. E. Sproull	O. W. Motz E. L. Hogan
J. D. Hoffman D. A. Bridges (*A. S. M. E.*)	W. T. Miller (*Non-Member*) A. A. Potter (*A. S. M. E.*)
W. G. Boales R. F. Connell	R. K. Milward L. J. Potvin (*Non-Member*)

CANDIDATES

WARD, H. H., Air Cond. & Htg. Engr., Meltzer Motors, Miami, Fla.

WHELAN, W. J., Purchaser & Estimator, Harrigan & Reid Co., Detroit, Mich.

WOLIN, M. W., Jr., Engr. & Draftsman, Typhoon Air Cond. Co., New York, N. Y.

REFERENCES

Proposers	*Seconders*
L. L. Kuempel	D. H. Friedman, Jr.
S. G. Pizie	W. R. Moore (*A. S. M. E.*)
F. J. Feely	E. H. Clark
W. G. Boales	R. K. Milward
W. R. Zuhlke	A. A. Adler
M. C. Giannini	M. M. Kahili (*Non-Member*)

Candidates Elected

In the past issues of the JOURNAL of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

MEMBERS

BATES, H. C., Chief of Fibre Prod. Lab., Corning Glass Works, Corning, N. Y.

BRATT, H. D., Individual Ownership, Temperature Control Systems, Grand Rapids, Mich.

CROSS, R. C., Fuel Engr., Battelle Memorial Inst., Columbus, Ohio.

DION, A. M., Air Cond. Engr., Sheldons, Ltd., Galt, Ont., Can.

GRIESS, P. G., Mech. Engr., Voorhees, Gmelin & Walker, Bogota, N. J.

HINES, J. C., Air Cond. Engr., R. B. Hayward Co., Chicago, Ill.

HOWARD, F. L., Chief Engr., Refrigeration & Air Cond. Inst., Chicago, Ill.

KEATING, A. J., Design Engr., Cooling & Air Cond. Corp., Chicago, Ill.

KERR, W. E., Engr. & Sales Repr., W. E. Kerr, Columbia, S. C.

KNAPP, A. E., Asst. Chief Engr., Kelvinator Corp., Detroit, Mich.

KUECHENBERG, W. A., Construction Engr., R. B. Hayward Co., Chicago, Ill.

KURTH, F. J., Tech. Director, Anemostat Corp. of America, New York, N. Y.

LEFEBVRE, E. J., Design Engr., Warden King, Ltd., Montreal, Que., Can.

LEUTHESSER, F. W., Jr., Vice Pres., National Metal Products Corp., Chicago, Ill.

LEWIS, G. M., Bldg. Engr., Simon J. Murphy Co., Detroit, Mich.

MABLEY, L. C., Sales Engr., Surface Combustion Co., Toledo, Ohio.

MATHEWSON, M., E. Engr., A. M. Kinney, Inc., Cincinnati, Ohio.

MATOUSEK, A. G., Engr., York Ice Machinery Corp., St. Louis, Mo.

MUELLER, J. E., Sales Engr., West Penn Power Co., Pittsburgh, Pa.

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WATTS, A. E., Mfr. of Boilers, Silent-Glow Oil Burners, Montreal, Que., Can.

WEINERT, F. C., Sales Engr., Chamberlin Metal Weather Strip Co., Detroit, Mich.

JUNIORS

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FINNERTY, J. A., Sales Engr., Herman Nelson Corp., New York, N. Y.

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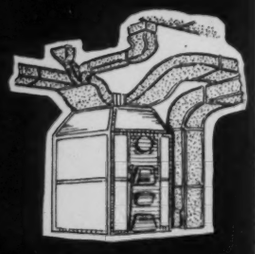

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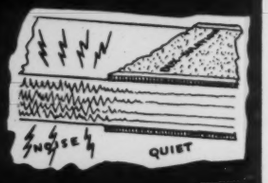

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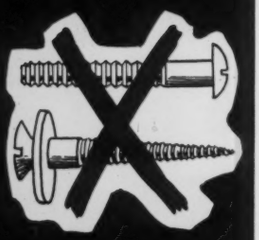

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Equipment Developments

For your convenience in obtaining more information about any of this equipment, see coupon on page 156. Add the new products and companies listed here to your Directory Section which you received in your January, 1937, HEATING, PIPING AND AIR CONDITIONING and thus keep your records of sources of supply up to date throughout the year . . . Single asterisk (*) indicates equipment not listed in Directory Section; double asterisk (**) equipment and manufacturer not listed.

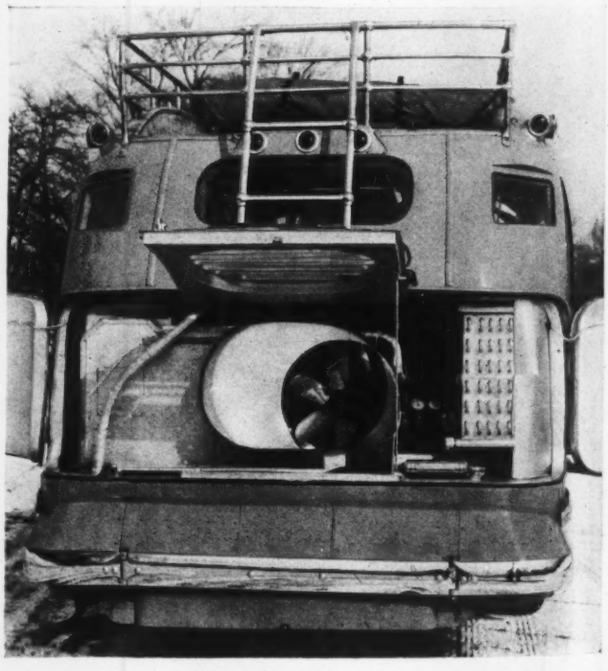
Air Conditioned Bus Tours South

No. 1196—An air conditioning unit developed for motor coaches is installed in a White Motor Co. research coach—named the "Lord Kelvin Coach"—which is now making a three month demonstration tour throughout the South.

Outside air, brought in and cooled by the especially designed unit, passes through ducts built into the luggage racks directly above the seats and running the length of the coach on both sides. The air is forced downward from these compartments onto a mushroom horizontal deflector which circulates the cooling air in such a manner as to envelop the passengers indirectly, avoid discomfort of cold air drafts. The outside air enters at the front of the body and is conveyed to the air conditioning unit at the rear, where it is filtered, cooled, and dehumidified.

The equipment operates continuously. Power for both refrigerant compressor and air circulation fan is accomplished through belt drive from a gasoline engine designated as the power unit, which operates continuously at constant speed. The loading of this engine varies with the demand for cooling. When the bus comfort temperature is satisfied, the thermostat breaks the circuit to liquid solenoid valve. This stops circulation of the refrigerant to the cooling coil. Since the refrigerant compressor operates continuously, it eventually pumps all the refrigerant back through the air cooled condenser to the liquid refrigerant receiver.

During this operation the suction pressure continues to drop and eventually reaches a stage where the low pressure cut-out closes the circuit to the bypass solenoid valve. This opens the bypass circuit, equalizing the head and suction pressure on the compressor, and accordingly reduces the load on the engine to a minimum. During this off cycle, the gasoline engine continues to operate at reduced load until cooling is again called for. Then the refrigeration cycle is automatically re-established.



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RHOADS TANNATE-ROCKWOOD DRIVES *for* VENTILATING *and* AIR CONDITIONING

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In Tannate Belts pliability, strength, grip and durability **are combined** in an unusual degree.

You can feel sure that the desired speeds and output from your fans and compressors will be maintained by using Tannate-Rockwood Short Center Motor Drives.

The motor is mounted on a rugged Rockwood Pivoted Motor Base, insuring maintenance of the proper belt tension, predetermined by accurate engineering design. The high efficiency of the drive depends also in large measure on the unusual power transmitting qualities of RHOADS TANNATE LEATHER BELTS.

A Tannate-Rockwood Drive may be placed on the floor, ceiling or side-wall, or on a special foundation. In any case you will have dependable and guaranteed results. This drive is also one of the most efficient methods available for difficult vertical drives.

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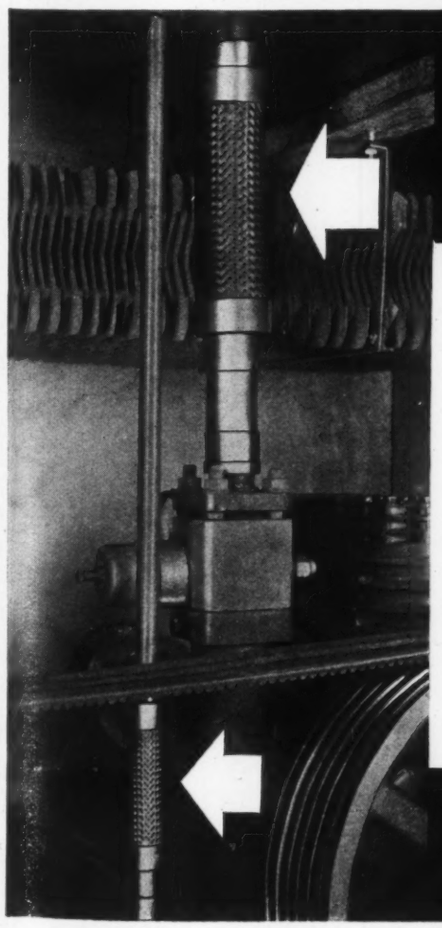

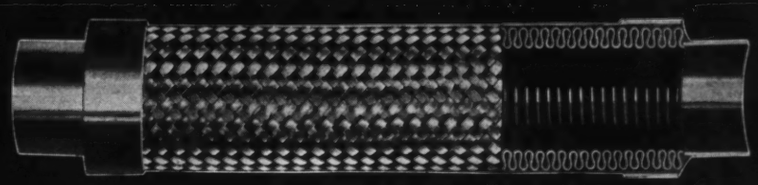

REX-WELD "Super-Service" VIBRATION ABSORBERS *for Air-Conditioning Machinery*

● Here's the ideal solution to the compressor noise problem! Installed in the suction and discharge lines, these Rex-Weld Flexible Metal Tubing Units effectively dampen the vibration and put an end to the annoying hammering—surely, safely and economically . . . Produced by an exclusive patented process which forms the flexible tubing corrugations without weakening the special, non-porous, non-crystallizing bronze alloy. Supplied complete with capped ends ready for sweating onto standard water tubing. Write for Bulletin RW-4 and Data Sheet. Send us your specifications for prices.

CHICAGO METAL HOSE CORPORATION
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for every
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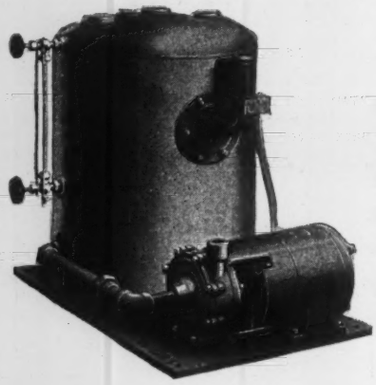

With
MONOBLOC
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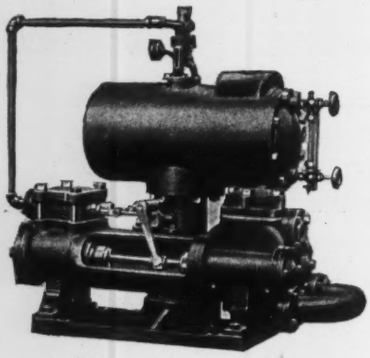
PUMPS
2000 to 100,000
sq. ft. radiation
Discharge pressures up to 53 lb.



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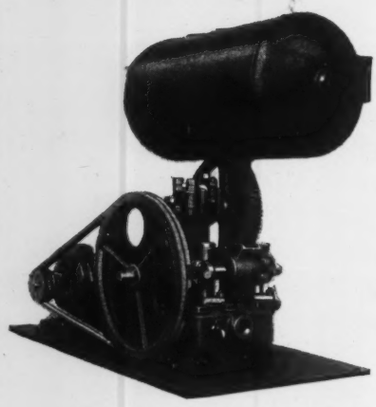

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1600 to 39,200
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Discharge pressures to 250 lb.

A COMPLETE LINE in each type provides exact selections without compromise ... and with Worthington's unsurpassed experience behind every unit.

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IC7-21

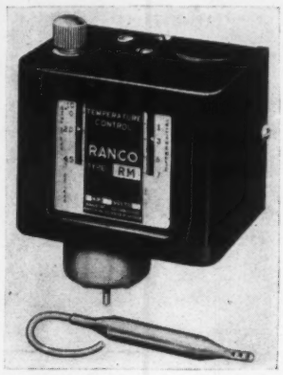
In the event of excessive head pressure, the high pressure cut-out automatically opens, causing the liquid solenoid valve to close and the cycle continues as described under thermostatic control.

The hand switch in the circuit of the bypass solenoid valve is manually closed prior to starting of the power unit in order to unload the compressor while cranking the engine. A second hand switch is provided for grounding the magneto when it is desired to stop the gasoline engine.

Production of the air conditioned units is scheduled for fall.—Kelvinator Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit, Mich.

Offers New Temperature, Pressure Control

No. 1197**—New temperature and pressure control is for use where temperature or pressure must be held to close limits to compensate for over-run caused by the sluggish characteristics of other units comprising the heating or cooling system.

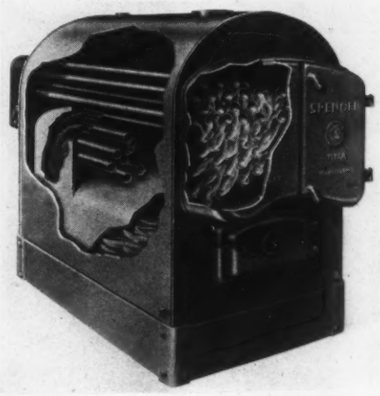


It is a combination of an auxiliary switch mechanism, operated by a temperature or pressure bellows, and a magnetically operated contactor. Auxiliary contacts require little power for operation, making possible a low differential. Main contacts are closed by magnet under heavy pressure with quick make and break providing a large current carrying capacity (1 hp, 110-220 volts a-c; 115-230 volts d-c). Available in temperature ranges from —20 F to +90 F. Low differential (1 F or less) adjustable up to 7 F. Construction is compact, overall dimensions being 2⅝ in. deep x 3$\frac{3}{32}$ in. wide x 2½ in. high.—Ranco, Inc., 601 W. 5th Ave., Columbus, Ohio.

Steel Heating Boiler Burns Any Fuel

No. 1198—New "Type A" steel heating boiler will burn any fuel, is offered in steam capacities of 1800 to 7260 ft. (*S.H.B.I.* rating).

The 3 in. tubes provide for long fire travel and are easily accessible for cleaning. Flanged supply outlet permits lower piping, consequently easier installation and minimum fittings. All tubes may be replaced without removal of the breeching. The doors, with machined ground contact surfaces, are of heavy cast iron construction and the flue cover doors are heavily ribbed. The tubes are cleaned through the front flue doors while in operation.



Fuel oil or gas firing may be installed at will and the boiler is adaptable for either hand or stoker firing. When solid fuel is used, grate sections are of heat resisting cast iron and sections of the same type are interchangeable.

Provision is made for a submerged water heater and either a storage type coil or instantaneous type water heating coil can be supplied.—Spencer Heaters Div., Lycoming Mfg. Co., 652 Oliver St., Williamsport, Pa.

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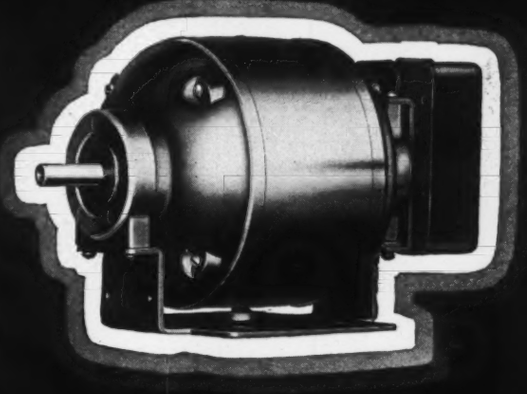
for *extra* QUIET OPERATION



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It's QUIET – both electrically and mechanically. Scientifically rubber mounted to dampen vibration and provide the utmost in Quiet Operation.

Available in both Split Phase and Condenser types – large and small sizes.

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A BETTER MOTOR

THAT'S WHAT GIVES YOU MANY MORE QUICK CLEAN CUTS Practically without Burr

The RIDGID Thin Tool-Steel Cutter Wheel

Not cut out of bar stock, this RIDGID Cutter Wheel is coined from tool-steel sheets, hammered, heat-treated and cast into a solid steel-bearing hub. This exclusive patented wheel gives you a durable cutting edge that means easier cuts and many more of them.

The cutter housing is steel re-inforced, guaranteed not to warp or break. The blade wheel cuts true.

RIDGID Heavy-Duty Cutter Wheel —for tough jobs and long service.

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RIDGID Tubing Cutter smooths the tube for soldering as it cuts, has integral reamer. Made in 2 sizes, for ⅛" tubing up.

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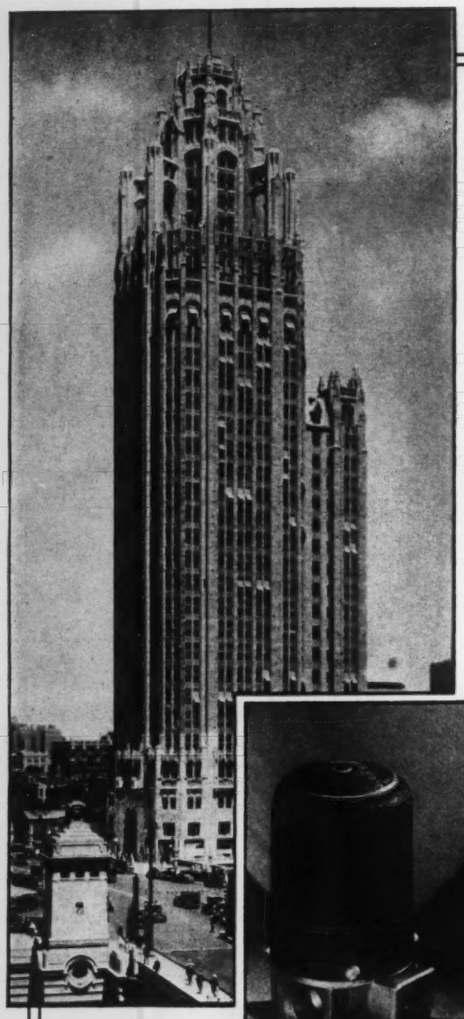
If you want to reduce your wrench repairs fully 75%, buy RIDGID Pipe Wrenches with unconditionally guaranteed housings.



400 A-P Solenoids Serve Tribune Tower

The task of air conditioning the famous Tribune Tower is performed by approximately four hundred individual room cooling units equipped with (A-P) Solenoid Refrigerant Valves.

The increasing regularity with which (A-P) Solenoids are selected as the refrigerant control units in air conditioning America's outstanding buildings, reflects a well-grounded recognition of the proven performance standards of (A-P) Valves.

A tip from experience — look to (A-P) to provide the maximum safety and performance on any air conditioning application.

Progressive jobbers everywhere stock (A-P) Controls!

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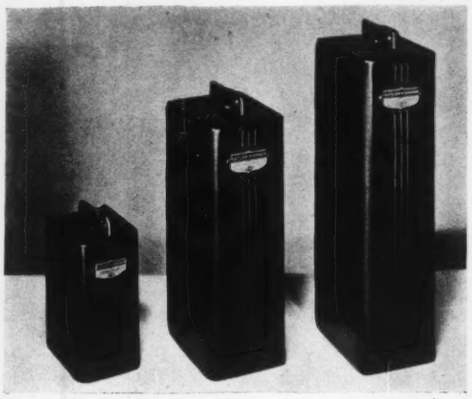
A-P MICROMETRIC PRECISION

Fractional Horsepower Drum Controllers Offered

No. 1199—New fractional horsepower drum controllers are for use with reversible a-c squirrel cage motors and reversing or non-reversing multi-speed squirrel cage motors driving machines where speed regulation is desired.



An "airstyled" removable case encloses a compact mechanism offering such features as insulated cylinder to which the segments are firmly secured without screws, non-stubbing, renewable contact fingers mounted in individual bakelite mouldings and easily accessible for wiring, positive action, self-indexing drum.—Cutler-Hammer, Inc., N. 12th St. & W. St. Paul Ave., Milwaukee, Wis.

Develops New Oil Chiller Equipment

No. 1200—Improved shell and tube oil chiller is expected by the manufacturer to facilitate the process of obtaining low temperatures required in multiple-stage chilling of low cold test oils, and to prove of practical value aiding rapid refrigeration in the oil industry, as well as in other industries where low temperatures are essential.

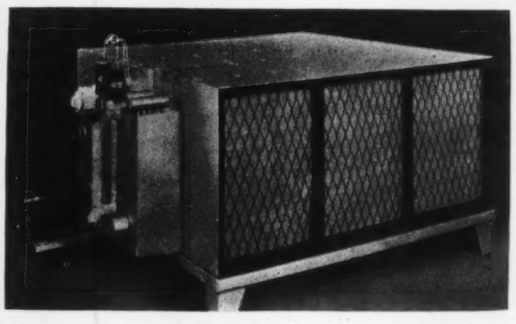
An eight shell unit of the new type is but 23.5 x 5 x 10 ft, when insulated, and has 1170 sq ft of effective chilling surface, whereas a 16-pipe section chiller unit requires a space, when insulated, of 51 x 3 x 12 ft, (without the motor) for the same amount of effective chilling surface.

Another feature is that loss of solvent through the gland is eliminated.

Scrapers of special design with sharpened blades remove all wax from the tube surface to assure high rate of heat transfer.—York Ice Machinery Corp., Roosevelt Ave., York, Pa.

Announces All-Season Conditioning Units

No. 1201—Line of "all-season" unit air conditioners built in 3 to 15 ton capacities for summer and winter air conditioning are factory engineered, ready to install. They are self-contained units cataloged and delivered on a package basis for combined air



filtering, cooling, heating and humidifying, or for heating or cooling only.

Coil surface is available for use with refrigerant, cold water, steam and hot water. Features include—states the maker—

"MIGHTY MIDGET"
Now Only $59.00

Cuts up to 18 gauge galvanized iron, sells at a price within the reach of anyone having sheet materials to cut. Other portable Unishears for 16 and 14 gauge, and stationary models with capacities up to ¼" boiler plate.

REDUCES SHOP COSTS

Equipment that will cut shop costs is worth looking into, these days! That's why it's time for *you* to investigate Stanley Unishears, the electrically-driven hand shears.

We're giving you the words of hundreds of heating contractors when we tell you that a Unishear will pay for itself in a short time, cutting sheet metal in the shop or on the job.

Operating at 2400 shear cuts a minute, a Unishear slices through ducts or sheets quickly and smoothly. Follows any pattern, around curves or angles. Leaves no rough edges; does not distort the metal. Speed, as you feed. Handles easier than a pair of snips.

You'll say the Unishear is one of the best investments you ever made, when you see the hours it saves; the better work it helps you do. Ask the Stanley distributor for a free demonstration, or write for details. Stanley Electric Tool Division, The Stanley Works, 102 Elm Street, New Britain, Conn.

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The use of low-pressure refrigerants in modern air conditioning applications has brought about revolutionary achievements in compressor design, as exemplified in the remarkable "Freon 12" Vilter Compressor. Through the unique application of a proven basic principle, Vilter engineers have produced a positive shaft seal and lower horsepower per ton. For appearance, performance and economy of operation, none can compare with the new Vilter "Freon 12" compressors. Write today for complete information.

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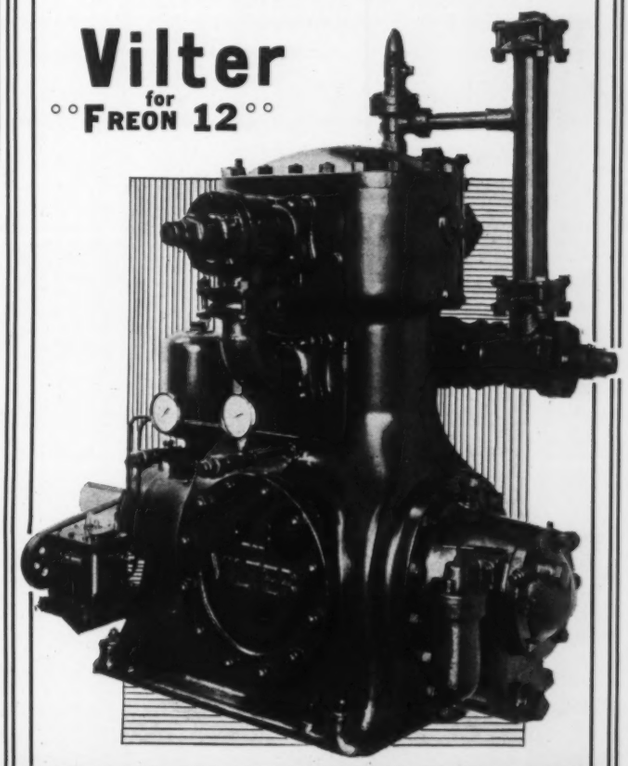
°°FREON 12°°



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"SINCE 1891"
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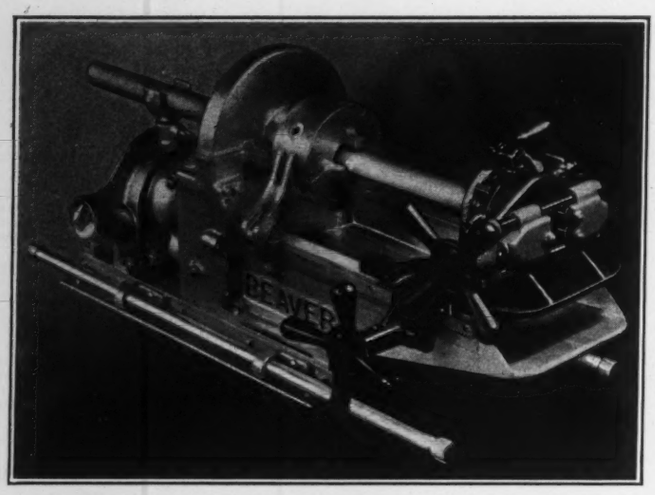
MILWAUKEE, WISCONSIN
REPRESENTATIVES IN PRINCIPAL CITIES



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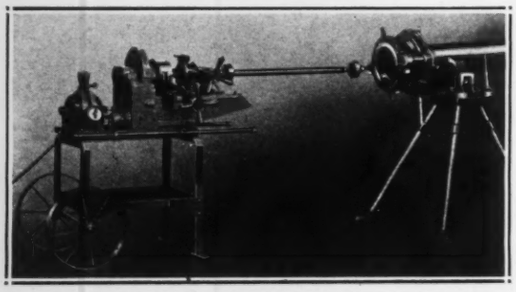
If so, you will appreciate the value of these three features of the Beaver Model-A Portable Pipe Machine.

1. Right-Handed
Conforming to standard machine tool design, the Model-A is right-handed like a lathe—with chuck to the left and tool-mounting and operating controls to the right. All controls in front—facing operator!

2. Full-Range
It covers the full range ⅛ to 2-inch instead of the usual range ½ to 2-inch. It cuts, threads, reams and chamfers nine sizes instead of six sizes. Those extra sizes, viz.; ⅛, ¼ and ⅜-inch are important in every industrial plant where small pipe is used for air-lines, lubricant lines, gases, etc.

3. Double-Duty
By virtue of its patented wheel-cutoff, the Model-A will both cut and thread solid round bars—for making threaded rods, bolts, pipe hangers, etc. This extremely important service feature cannot be had efficiently from machines equipped with knife-type cutoff.

When you buy a pipe machine demand these three advantages. In the meantime write us for complete descriptive literature of the Model-A Beaver Special—priced at $295.00 complete, ⅛ to 2-inch.



With drive shaft and geared tools, the BEAVER Model A will cut and thread 2½ to 12 inch pipe.

BEAVER PIPE TOOLS

437 KEENEY AVE. WARREN, OHIO

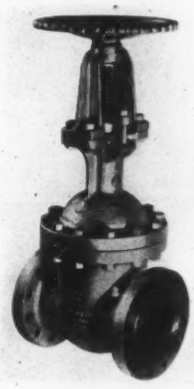
lighter weight per capacity; compact size per capacity; all-copper heat transfer surface manifolded for proper distribution; easy accessibility to entire interior of cabinet: interchangeable parts for floor or ceiling mountings; rust-proof construction throughout.

Direct expansion coils are equipped with high capacity thermostatic expansion valves accessibly located for accurate refrigerant control in accordance with varying temperature differentials encountered as air progresses through coil.

Humidification is by means of a self-cleaning nozzle which ejects a fog into the air stream, and operation can be controlled either manually or automatically. Provision is made for adding the humidifying unit at any time if it is not required on the original installation.

Filter section containing double filter of removable type is protected with access plates for easy replacement of filters when necessary.—Fedders Mfg. Co., Inc., 57 Tonawanda St., Buffalo, N. Y.

Steel Gate Valves Have Many Features



No. 1202—Heavy wall thicknesses, deep stuffing boxes, oversize stems, streamline flow and steel handwheels with fluted non-slip rims, are some of the features included in new steel gate valves now available for 150 and 300 lb steam pressure.

Made of carbon molybdenum steel (identification symbol C-1) to *A.S.T.M.* specification A 157-36, they meet the requirements of all applicable specifications, the maker says. Illustration shown is from an unretouched photograph of one of the 300 lb steel gate valves—Walworth Co., 60 E. 42nd St., New York, N. Y.

Portable Room Conditioner Tested in Brazil

No. 1203—Engineers gained a "climatic year" by testing new portable summer air conditioner in Brazil during January and February, hot months of the South American country.

The air cooled model, embodying several new features, was made to sell for $395 f.o.b. factory; it is completely factory assembled.

Warm outside air is drawn into the unit through an adjustable window duct, made to fit any size opening. The air then passes over cooling coils and is filtered, cleaned and circulated over the room. A feature noted in the Brazilian experiment was the low power cost in relation to the good performance of the machine, states the maker. One reason is because of sub-cooling coils that serve as an economizer. Water is drawn from the air and drips to a drain pan in which is located sub-freezing coils. This cools the refrigerant "Freon," adding to the efficiency of the equipment.

The unit is lined with

Standardization does not stop with the attractive, warmly colored terra cotta tile front and uniform architectural style of all stores of S. H. Kress & Co. For Sarco Heating Specialties have also been the Kress standard for many years.

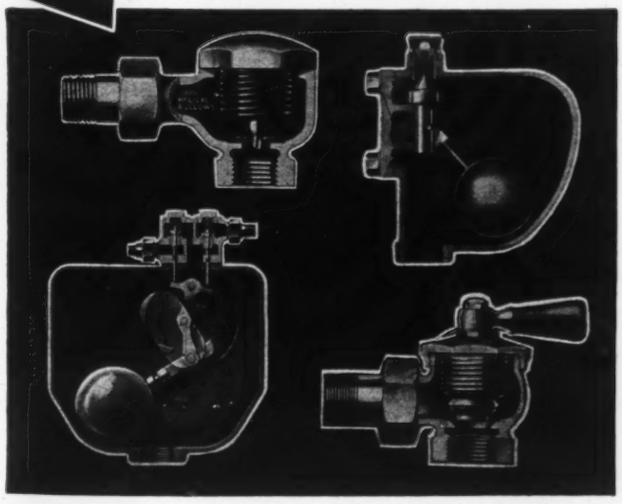
A product must be good when shrewd buyers like these continue to buy it year after year as new stores are erected.

Sarco have been making steam specialties for a quarter of a century and should know how to build a good heating system.

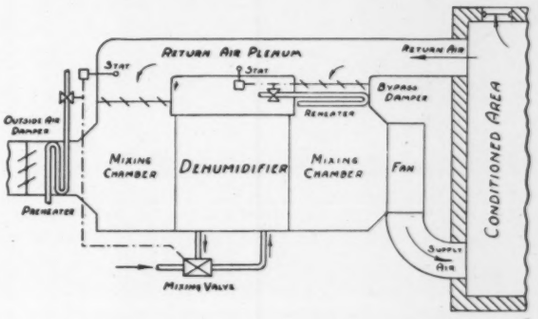
It may be to your advantage to learn more about the Sarco. Why not write for Catalog O-45?

SARCO COMPANY, Inc.
183 MADISON AVE. NEW YORK, N. Y.

BRANCHES IN PRINCIPAL CITIES
SARCO CANADA LTD., FEDERAL BLDG., TORONTO, ONT., CANADA



AN AIR CONDITIONING SYSTEM OF UNIVERSAL APPLICATION



AUDITORIUM Conditioning Corporation in this and succeeding months will bring to your attention outstanding types of air conditioning systems (*) for **HUMAN COMFORT.**

The system illustrated is found in most of the prominent air conditioning installations in the world. Leading architects and engineers consider it **STANDARD.** Its pre-eminence is based upon outstanding features, among which are—

- √ Maximum efficiency and economy under all load conditions.
- √ Speedy cooling down.
- √ Simplicity and flexibility of control.
- √ Temperature and humidity always as desired regardless of occupancy or load.
- √ An ample supply of fresh air for ventilation requirements.
- √ No streaks or drafts.
- √ Year 'round operation — summer and winter.

Our Licensees will be glad to consult with you regarding the engineering features of this system and also tell you about many other Auditorium Systems. (*)

American Blower Corp.
Detroit, Mich.

Buffalo Forge Co.
Buffalo, N. Y.

Carrier Corp.
Newark, N. J.

Frick Co.
Waynesboro, Pa.

General Electric Co.
Schenectady, N. Y.

J. O. Ross Engineering Corp.
New York, N. Y.

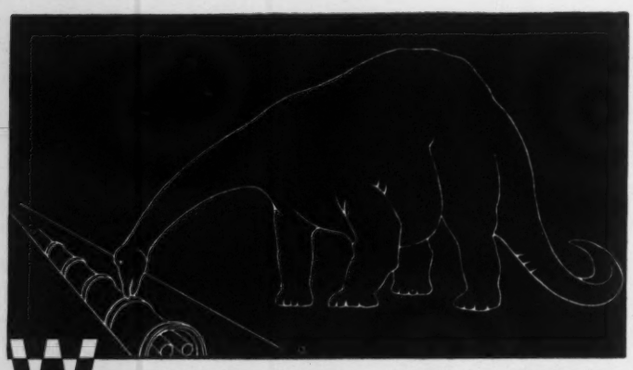
The Cooling & Air Conditioning Corp.
Division of B. F. Sturtevant Co.
Hyde Park, Boston, Mass.

York Ice Machinery Corp.
York, Pa.

(*) *Auditorium Air Conditioning Systems are covered by many issued patents and pending applications.*

AUDITORIUM CONDITIONING CORPORATION

New York Office—17 East 42nd Street . New York



WHAT'S THIS?
SAID THE DINOSAUR

AN underground steam conduit is just about as unfamiliar to most people today as it would have been to Mr. O. U. Dinosaurus 50 million years ago. Here you see the old boy giving it the once-over. If it's Ric-wiL, it will stand the closest inspection he, or you, can give it. If you had a neck as long as his, you could poke your nose down in the ditch and see what it's all about. Even if you don't, it will pay you to analyze in advance the Underground Conduit you invest in, from EVERY angle. When you do that you'll see why Ric-wiL is considered the FOREMOST system for EFFECTIVE steam protection.

All basic qualities—water repellence, heat retention, underdrainage, alignment, ease of assembly—are fully met in Ric-wiL systems. Ric-wiL keeps steam lines over 90% efficient. Various designs in Tile or Cast Iron to meet all conditions—famous Dry-paC Waterproof Asbestos Insulation. Ask for complete catalog with Dry-paC sample for testing.

The Ric-wiL Co. Union Trust Bldg. Cleveland, Ohio
New York San Francisco Chicago
Agents in principal cities.

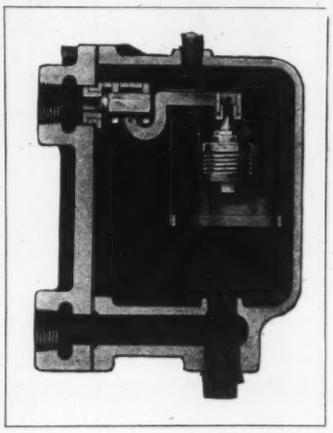
Avoid makeshift imitations which may look and sound like Ric-wiL. Only GENUINE Ric-wiL gives Ric-wiL protection.

REGISTERED IN U. S. PATENT OFFICE
RIC-WIL
CONDUIT SYSTEMS FOR
UNDERGROUND STEAM PIPES

rock wool insulation for deadening sound. The fan discharges into a sound absorbing chamber, also insulated. All moving parts are mounted on springs for vibration elimination.—Carrier Corp., 850 Frelinghuysen Ave., Newark, N. J.

Completes Line With Inverted Bucket Trap

No. 1204—New inverted bucket trap, which completes extensive steam trap line can be furnished with integral air bypass carried inside the bucket and consisting of a regular thermostatic trap element of the balanced pressure type. This air bypass never requires setting and operates satisfactorily regardless of pressure fluctuations in the trap, says the maker.



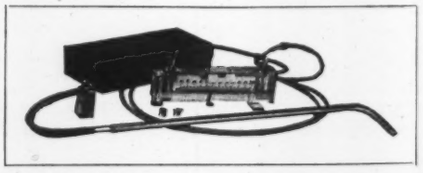
Bodies are of ample size and heavy design. Traps are available in ½ to 2 in. sizes with bodies of cast iron for 125 lb, semi-steel to 250 lb, and cast steel up to 500 lb steam pressure. Valve heads and seats in all types are of stainless steel.—Sarco Co., Inc., 183 Madison Avenue, New York, N. Y.

Draft and Air Filter Gauges Improved

No. 1205—New air meters, draft gauges, and air filter gauges, which incorporate several new advantages over earlier types of instruments of this sort, as well as a new pitot static head for use with the portable types, were announced last month.

These new instruments are constructed of a clear bakelite material, and are provided with accurate, quick, and easy means for leveling before taking readings. The portable instrument with its rubber tubing and other fittings, including the bottle of spare oil is conveniently contained in a carrying case. Various types and ranges are available covering velocities as low as 400 and as high as 8000 fpm, or vacuums or pressures as low as 0.005 in. of water up to 4 in. of water.

A unique feature of the instruments is the chromium plated and polished graduated scale which is carried on the back of the instrument and viewed through the transparent bakelite. This polished scale reflects an image of the liquid column, and the reading of the instrument in use, therefore, consists of lining up the actual liquid column, the graduated scale (which has a zero adjustment feature), and the reflected image of the liquid column which appears to be beyond the scale. Accurate readings are obtained because parallax errors are eliminated.



The pitot static head is for use either with the instrument described above, or with other instruments designed for similar purposes. It is possible to obtain accurate results even when it is held in such a way that the air stream it is facing is coming from a direction as much as 15 angular degrees from any side of the center line of the head. The illustration shows the portable type "AM/2" with the pitot static head, carrying case, and other accessories.—Julien P. Friez & Sons, Inc., 4 N. Central Ave., Baltimore, Md.

IT'S *Your* RESPONSIBILITY TO SELECT THE PIPE

It's a good guess that nine out of every ten jobs are steel pipe jobs in which the selection of the pipe is left to you. Thus, it becomes your responsibility to select pipe that will give your customer full dollar value and at the same time insure profit for you. • That's why many engineers and contractors have discontinued the use of "any steel pipe." They insist on Fretz-Moon Pipe — because it is sound in every respect and because it permits them to use pipe of unusually high quality at no increase in price. • They know, too, that because of the exclusive "continuous process" of manufacture, all uncertainty and guess work is eliminated — and the finished pipe is exceedingly uniform in size, weld, ductility, cleanliness, threading and galvanizing — thus making possible economies in installation. • Ask your jobber for Fretz-Moon Pipe. If he cannot supply you, write us.

FRETZ-MOON TUBE CO., INC.
BUTLER • PENNA.

TRY FRETZ-MOON PIPE
—AND KNOW THE DIFFERENCE

GOOD BUILDINGS *Deserve* GOOD HEATING...

Into the making of a good building go knowledge, judgment and skill. From it comes services. In use, the building is intended to distribute the values the Architect, Engineer and Contractor have created. Only the comfortable building fully satisfies this intention.

That is the reason a good building—whether commercial, industrial, residential, institutional or educational—deserves good heating.

Dunham Differential Heating keeps good buildings comfortable. It distributes subatmospheric steam ("cool" steam) at proper rates to maintain an equable building environment, the more pleasing by reason of the normally low temperature of radiators which WARM rather than scorch the air they keep continuously in gentle motion. It simplifies the problem of winter air conditioning.

For owners, lower fuel consumption is the satisfying paradox of greater building comfort. For Consulting Engineers and Heating Contractors, good heating service is an enduring comfort proof of competent services. C. A. Dunham Company, 450 East Ohio Street, Chicago.

DUNHAM *Differential* HEATING

Before and during building completion and later, — on "operations," the Consulting Engineer should be an advisor regarding mechanical equipment.

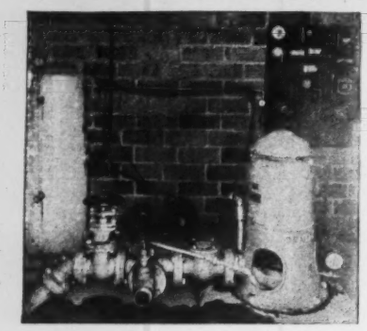
Gulf Building
Houston, Texas

Architect:
Alfred C. Finn

Consulting Architects:
Kenneth Franzheim
J. E. R. Carpenter

Heating Contractor:
C. Wallace

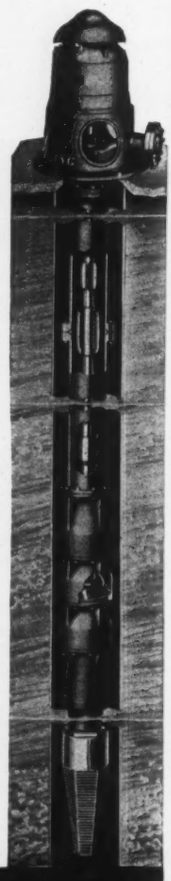
ECONOMICAL
Summer Air Conditioning
WITH DEMING TURBINE PUMPS



In most cases... where well water is available... the total cost of a well, a Deming Turbine Pump plus its operation and maintenance is less by far *than any other type of evaporative cooling equipment.*

Deming Turbine Pumps offer an effective and economical solution to summer air conditioning problems where well water is available as the cooling agent. Among the many noteworthy features of Deming Deep Well Turbine Pumps are...

1. Water lubrication which continuously floods and positively lubricates all bearings. (There is no oil to contaminate water used for any purpose... **2.** Highest quality "cutless" rubber used in bearings resists wear and contributes to maximum performance and dependable operation of Deming Turbine Pumps... **3.** Impeller clearance is easily adjustable from surface for maintaining original efficiency even after extended periods of hard service... **4.** Solid stainless steel shaft used in bowls and shaft with stainless steel sleeves in the column... Full weight copper-bearing pipe is used for column.

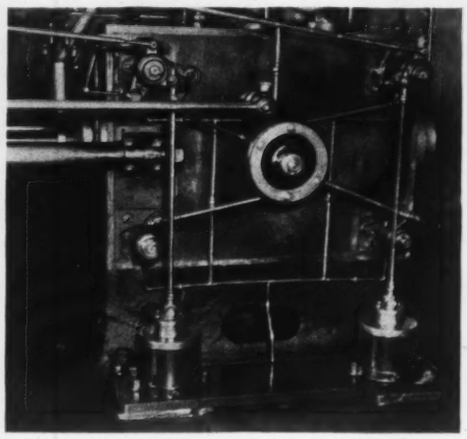


•Deming Deep Well Turbine Pumps are built to operate in wells of 5 5/8" and larger inside diameter and range in capacity from 25 to 2000 gals. per min. Write for Bulletin.

DEMING
Pumps
THE DEMING CO. • SALEM, OHIO

Compressor Valve Latch Plates Hard-Faced

No. 1206—Faulty valve operation and over-all inefficiency result if the latch plates on the valve gear of this ammonia compressor become rounded from wear. The life of these latch plates was formerly one month, but after being hard-faced on the



wearing edges with an alloy of cobalt, chromium and tungsten, they last for over 3 years. On old equipment such as this compressor, parts for which are out of stock, rebuilding with the oxy-acetylene blowpipe greatly reduces maintenance and replacement expenses, according to Haynes Stellite Co., 205 E. 42nd St., New York, N. Y.

Offers Thermostatic Tri-Metal

No. 1207**—A new type of thermostatic metal is known as "Tri-Ply" and differs from the conventional bi-metal in that a third element has been placed between the active and inactive members. The addition of this third element, says the maker, has resulted in this thermostatic metal having smoother operation and more uniform movement throughout its operating range, which is from 0 to 1200 F. It can be furnished in any shape or in strip form.—Laminated Metals Corp., 775 Eddy St., Providence, R. I.

Protects Air Conditioning Equipment

No. 1208—Extremely fine metallic copper can be applied to iron, steel and wood to provide an effective protective coating. Known as "Coppercote," this new method has solved many corrosion problems, according to the manufacturer, and is adapted to protection of air conditioning equipment.

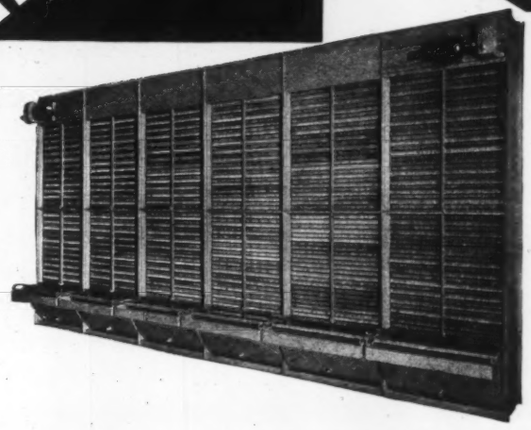
"Coppercote," when applied by a brush or spray, spreads automatically to form an impervious coating. A turbulent action takes place when the coating is applied causing it to spread automatically and work its way into the pores of the coated surface. This action throws off air bubbles and closes pin holes. A definite stratification of the metallic copper and its vehicle occurs causing the copper particles to combine and adhere closely to each other while the vehicle rises and forms a film which contains a color pigment. It contains an insulating element to prevent electrolytic action.

It may be applied to the surfaces of air conditioning apparatus, such as the lining for spray chambers and drip pans where there is a continuous presence of moisture.—American Coppercote, Inc., 480 Lexington Ave., New York, N. Y.

Large Electric Pliers For Sweating Pipe

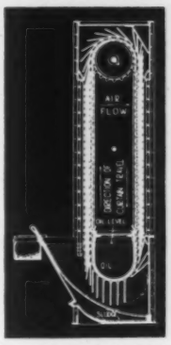
No. 1209—Supplementing the electric pliers introduced several months ago as a novel type of electrically heated soldering tool for small work, larger size pliers are now offered for soldering lugs up to 1050 amp. or sweating pipe fittings up to 2½ in. diameter, under continuous operation. When used intermittently, they will handle pipe or fittings up to 4 in. diameter.

MAINTENANCE FREE AUTOMATIC AIR FILTRATION



Uniform air-flow assured by self-cleaning louvre plates.

The "DOUBLE-DUTY" Air Filter has been acclaimed an outstanding automatic, self-cleaning filter for heavy duty work by leading manufacturers and mills thru-out the country. Simplicity in construction and design practically eliminates all mechanical attention. "DOUBLE DUTY" filtration only requires servicing to the extent of oil replenishment every three to six months, depending on the dust contents of the air in each individual case.



Cross-section view of filter, showing simplicity of self-cleaning operation.

"INDEPENDENT"

INDEPENDENT AIR FILTER COMPANY
215 WEST OHIO STREET, CHICAGO, ILLINOIS

ARCO COPPER-TO-COPPER

Patent Nos.: 2,025,973—2,005,969 and 2,002,470

SAFE WITH ANY REFRIGERANT

NON POROUS

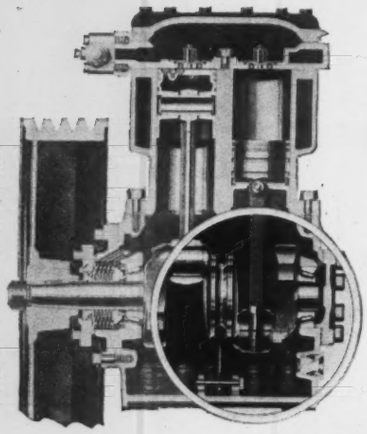
FOR AIR CONDITIONING & REFRIGERATION

NEEDS NO TINNING—Arco Copper-to-Copper makes the ideal joint. Safe against leaks of any kind. Safe against vibration. Absolutely non-porous every place in the line. Smooth inside . . . with Wrought Copper Full Flow Fittings...reduces friction loss. Easier to install, resists corrosion, lasts longer, costs less in the long run. Great variety of fittings from ¼" to 4".

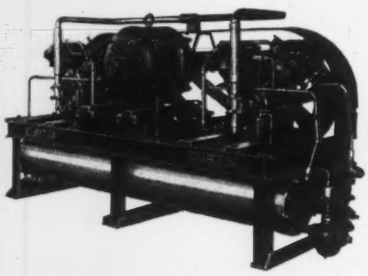
ARCO PIPE AND FITTINGS DIVISION
AMERICAN RADIATOR COMPANY
DIVISION OF AMERICAN RADIATOR & STANDARD SANITARY CORPORATION
40 West 40th Street, New York, N. Y.

ARCO
Full Flow WROUGHT
COPPER FITTINGS & PIPE
STOCKED BY JOBBERS EVERYWHERE

CURTIS
Condensing Units



"CENTRO-RING" LUBRICATION—
—only
CURTIS COMPRESSORS
have it



THERE'S just one moving part in Curtis "Centro-Ring" lubrication—a simple, free running ring that picks up oil from the crankcase and distributes it by centrifugal force under positive, uniform pressure to all points of lubrication—no gears—no plungers—nothing to get out of order or require service.

Engineering superiorities of this kind have built the Curtis reputation for reliability, long operating life and low cost of upkeep.

The Curtis line contains 41 air-cooled units, from 1/6 HP to 5 HP, inc., and 45 water-cooled units from 1/3 HP to 30 HP — a size and type to balance most any installation you might make. Curtis has built condensing units for 15 years and has made engineering equipment for 82 years — a record of success that in itself is sufficient warranty of the high-quality of Curtis units of today.

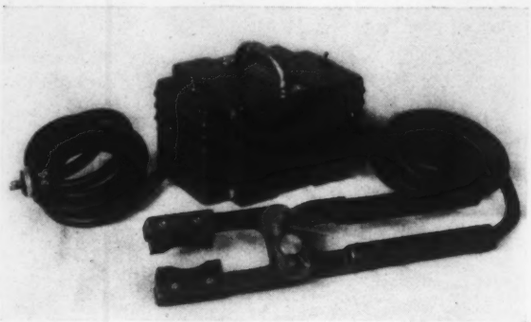
CURTIS Engineering Refinements

- Centro-Ring — one piece, positive pressure lubrication
- Timken Tapered Roller Bearings
- V-type radial compressor
- Water-jacketed compressor heads and cylinders
- Built-in oil separator with automatic return
- Drop forged, heat treated crankshafts and rods
- Balanced syphon, bellows seal
- Automatic water valves

Represented in Canada by
Canadian Curtis Refrigeration Co., Ltd.
20 George St., Hamilton, Ont., Can.

CURTIS REFRIGERATING MACHINE CO.
"Builders of Condensing Units since 1922"
1950 Kienlen Ave., St. Louis, U. S. A.

According to the maker, they sweat joints without unsweating adjacent connections, hold work firmly while soldering, heat the work evenly from both sides, and eliminate open flame hazards.—Ideal Commutator Dresser Co., 1537 Park Ave., Sycamore, Ill.



Fractional Hp Motors Have Resilient Mounting

No. 1210—Annular type, resilient mounted motors manufactured in sizes from ⅛ to 1 hp represent a mechanical modification to the maker's line of fractional horsepower motors. The mountings are available on repulsion start induction, split-phase, capacitor and polyphase types. This mounting has the advantage in that there is less vibration and hence less noise, since the rubber mounting supports which absorb the vibrations are as close to the shaft center as possible.

To meet the requirements of manufacturers of small belt driven appliances, these motors are also available with belt tightener bases so designed that the belt tension increases with the load to prevent belt slippage.—Wagner Electric Corp., 6400 Plymouth Ave., St. Louis, Mo.

Wide Range Features New Controllers

No. 1211—New non-indicating "wide range" temperature and pressure controller operates on 15 lb air pressure and has a throttling range of from 1 to 15 per cent. The temperature type has a graduated range of from —30 to 500 F, and the pressure type a graduated range of from 15 to 500 lb. Where required the controllers are supplied with balanced capillary tubing overcoming room temperature disturbance. Furnished with vapor tension bulbs with selection of fittings, connections and separable sockets to fit the individual job.—American Schaeffer & Budenberg Div., Consolidated Ashcroft Hancock Co., Inc., 11 Elias St., Bridgeport, Conn.

Thermostat Can Be Tuned to Job

No. 1212—New room thermostat named the "Metrotherm" is of low thermal inertia and contains a small rheostat which

Yet There's No Damage

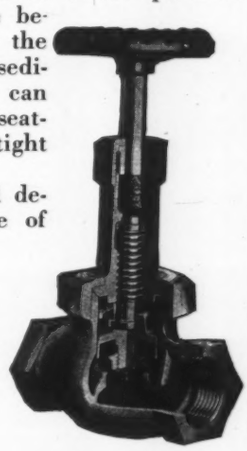
A smashing blow with a hammer won't injure Fairbanks Asbestos Disc Rings—a more severe test than any ring gets in service. They withstand brutal punishment that would disintegrate ordinary rings because they are reinforced with genuine long-fibred asbestos.

Another thing which prolongs the life of these rings is the design of the Fairbanks Disc Nut which throttles the action of the stream against the disc.

Fairbanks Disc Rings give equally long and efficient service with steam, hot water or cold water, eliminating the necessity of stocking several types.

The slide-on disc holder drops off when valve is taken apart, making it easy to change the disc. Perfect alignment is assured by the radial seat between body and bonnet when the valve is assembled.

Stuffing box can be packed under pressure without danger of leakage because the seat is above the threads on stem where no sediment or foreign matter can reach and pit or score the seating surfaces or prevent tight closing.

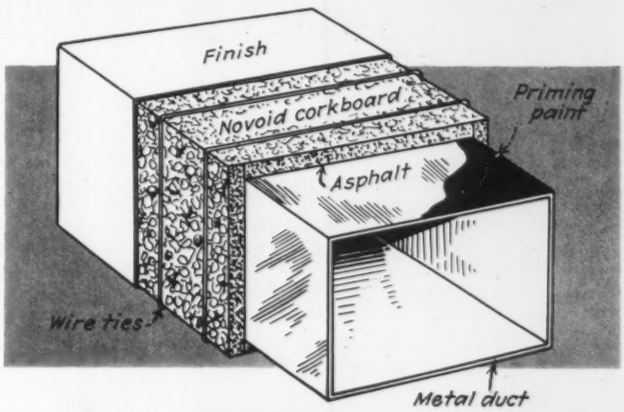


Write for Catalog No. 21 describing our complete line of bronze and iron valves.

The Fairbanks Company
Valves, Dart Unions, Hand Trucks and Wheelbarrows
396 Lafayette St., New York, N. Y.
Boston, Pittsburgh—Distributors in Principal Cities
Factories: Binghamton, N. Y., Rome, Ga.

Fairbanks
Renewable Valves

HEAT STAYS OUT COLD STAYS IN



when NOVOID *guards air ducts*

INSULATE air-conditioning ducts with Novoid Corkboard, and you have a heat resisting insulation that keeps heat out and cold in. Because of its unique cell structure, this efficient material not only resists heat, but also prevents condensation. It helps assure accurate control of temperature and humidity, and lowers operating costs.

Novoid Corkboard is light in weight, structurally strong, and may be easily and quickly erected on round, square, or rectangular air ducts. It is equally effective as an insulation for air-washing and cooling rooms, and prevents costly infiltration of warm air. For more complete information, mail the coupon, or write Cork Import Corporation, 330 West 42nd Street, New York City.

BRANCH OFFICES: Chicago, 400 West Madison Street; Boston, 27 Haymarket Square; Philadelphia, 1524 Chestnut Street.

NOVOID CORKBOARD

MAIL COUPON FOR FULL DETAILS

Cork Import Corporation
330 West 42nd Street, New York City

Please send me complete details of Novoid Corkboard and its application to air ducts.

Name .....

Street .....

City and State .....

allows for adjusting the degree of heat acceleration to the load characteristics, and to the type of installation to be controlled. In this way, the thermostat can be "tuned" to the individual installation to obtain maximum sensitivity and quick response, says the maker.

Design is simple, with all outer gadgets eliminated. Two slender pointers indicate the temperature setting and the actual room temperature on a modern dial. Finished in dull silver and chrome. Locking device standard. Available with timer. Night switch available at no extra cost.—General Controls Co., 267 5th Ave., New York, N. Y., or 1370 Harrison St., San Francisco, Calif.

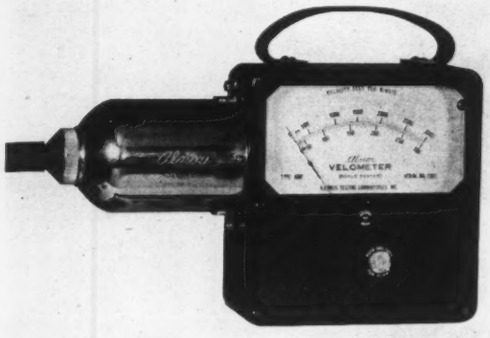
Announces Blower for Forced Draft Service

No. 1213—By a two stage combination, the pressures (6 or 7 in. w.g.) and volumes required for forced draft and similar work are obtained at low, quiet, constant speeds, regulation of the air volume being secured by built-in control and redirecting vanes, with a new blower. The units are of simple construction and have the driving motor self-contained; they are built for horizontal and vertical mounting.

The units consist of a constant speed electric motor, with double extended shaft; two propeller type fan wheels, one mounted on each of the two extended shafts; an integral damper control and air re-directing mechanism built in between the two propeller fan wheels. Applications include forced draft service for either hand fired boilers, stoker equipped boilers, oil or gas fired boilers, or for the secondary air supply of pulverized fuel burning boilers.—L. J. Wing Mfg. Co., 14th St. & 7th Ave., New York, N. Y.

Filter Extends Use of Air Velocity Meter

No. 1214—The illustration shows the "Alnor Velometer" (an instantaneous direct reading air velocity meter) with a new filter



attachment to permit use of this instrument in dust laden atmospheres either of the magnetic type dust or ordinary dust.—Illinois Testing Laboratories, Inc., 420 N. LaSalle St., Chicago, Ill.

Correction for March Summary Sheets

In putting pages 179 and 180 of the March, 1937, issue on the press, brackets were dropped from two of the heat transfer formulas shown on the "summary sheets" reproduced on those pages. The two formulas should read:

$$M = \frac{UA}{Gs}\left(1 + \frac{Gs}{G's'}\right) \quad [3]$$

$$M = \frac{UA}{Gs}\left(1 - \frac{Gs}{G's'}\right) \quad [4]$$

Typical industrial installation — one of several Marley forced draft towers purchased by a large food processing organization.
MORE water is cooled by Marley water cooling systems than by any other kind -- because Marley design and Marley performance provide maximum cooling efficiency.
THE MARLEY COMPANY
KANSAS CITY, MISSOURI
MARLEY
WATER COOLING SYSTEMS

CELOTEX
Reg. U. S. Pat. Off.

Solves Insulating Problems Quickly and Effectively

Because...

Celotex Insulating Boards have a thermal conductivity of only 0.33 Btu. per inch thickness . . .

And these large, lightweight, rigid boards cut easily with ordinary tools, go up fast, fit tight—and give lasting satisfaction.

Thus—Celotex-insulated ducts permit the maintenance of the desired temperatures economically.

For insulating cold air ducts, specify Celotex VLTI — the vaporproofed low temperature insulating blocks that are *completely* sealed against vapor penetration at the factory by a special wrapping and coating.

Celotex Insulating Cane Fibre Board and VLTI are protected against termites and dry rot by the exclusive Ferox Process (patented). Write us today for all the facts about these widely used insulations.

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CELOTEX
BRAND—INSULATING CANE BOARD
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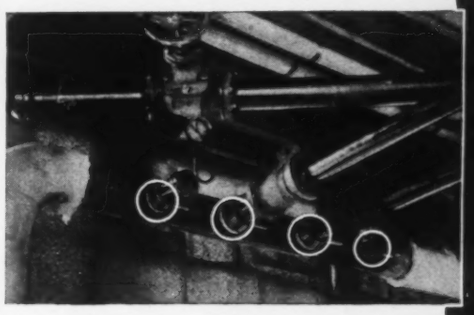
World's Largest Manufacturer of Structural Insulation

WeldOlets and ThredOlets
-the Perfect Welding Fittings
for Branch Line Take-offs



Among fittings for welded pipe installations, WeldOlets and ThredOlets are unique.

There is no other welding fitting that eliminates all cutting, threading, forming and fitting of the main pipe and branch—

that permits take-offs to be made with the main pipe in position—

that eliminates all templets—

that provides full pipe strength at the junction after the hole is cut in the main line—

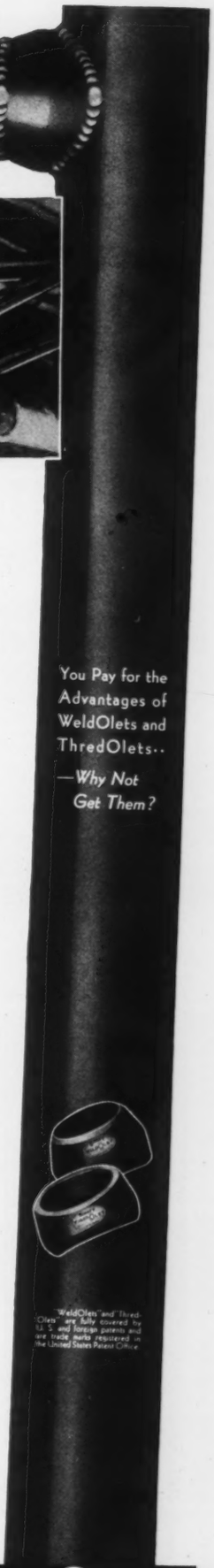
that results in an absolutely leak-proof joint for working under high pressures and high temperatures.

Yet with all these advantages, WeldOlets and ThredOlets cost no more (in many cases less) than ordinary fittings and considerably less to install. They are available from stock with outlet sizes from ½" to 12" for all standard size pipe.

Write today for your copy of Bulletin WT21 telling the whole WeldOlet-ThredOlet story.

BONNEY
FORGE & TOOL WORKS
Forged Fittings Division
Cedar & Meadow Sts.,
Allentown, Pa.

Carried in Stock by Leading Distributors


You Pay for the Advantages of WeldOlets and ThredOlets..
—Why Not Get Them?
"WeldOlets" and "ThredOlets" are fully covered by U.S. and foreign patents and are trade marks registered in the United States Patent Office

WELDOLETS·THREDOLETS

Question on Heating Bunkhouse Answered

THIS is an ordinary frame construction building to be used as a dormitory. It is unlikely that any scheme for heating except one having individual controllable radiators in the rooms would be satisfactory.

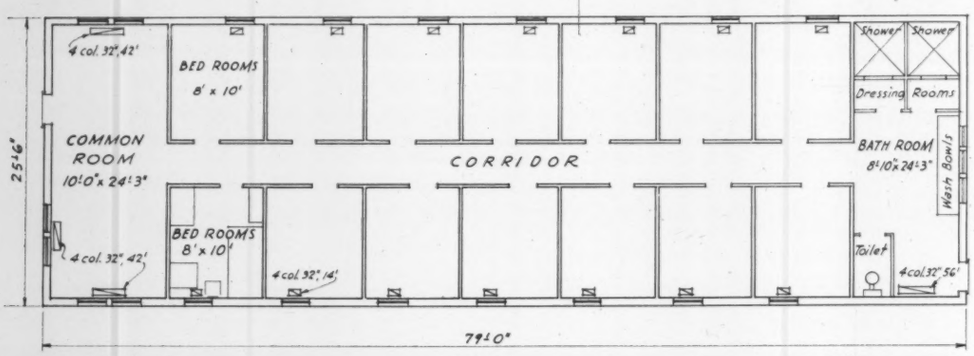
The average minimum outdoor temperature is not a sufficient criterion for design of the capacity of a heating system, since in making up this average figure it may be assumed pretty safely that no account was taken of the duration of each minimum temperature which was used in computing the average. Thus, a minimum average might be made up from the lowest temperature observed on one day in each month during the heating season, and it might be the 8 deg above zero which the question indicates. The *A. S. H. V. E. Guide* suggests a design temperature not more than 15 F above the lowest officially recorded temperature at any time during the preceding 10 years for the locality. Therefore, the amount of heat transmitting surface given should not be reduced.

There are 28 occupants. Suppose that within 10 minutes each

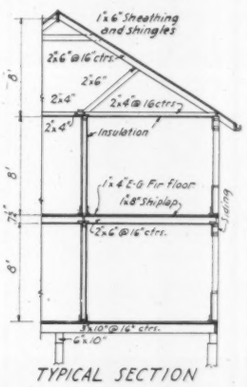
The Question

Herewith is the plan and section of a new bunk or rooming house for the employees of an industrial plant. I would like some information concerning the heating of this building.

The average minimum temperature during the last ten years in this locality has been 8 deg above zero. A plentiful supply of steam is available from a 2 in. pipe carrying 12 lb pressure, which is located 60 ft away from the new building. The pressure on the heating system is not to exceed 2 lb ga. Construction of the building is 2 in. by 6 in. studding on all outside walls, clapboards, paper and siding on outside, ⅜ in. insulation on inside and ceilings, common rooms being on the exposed or weather side of the building. The radiation has been figured and spotted, there being 378 sq ft on each of the two floors. The amount of radiation in each of the two common rooms seems to me to be excessive.



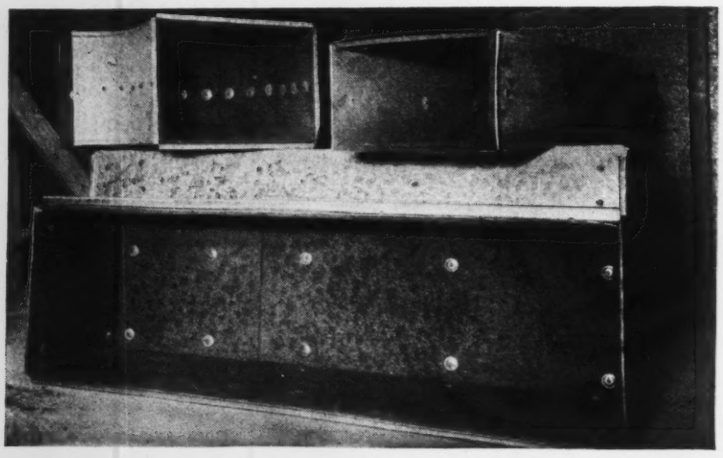
GROUND FLOOR PLAN



TYPICAL SECTION

LINING DUCTS for Sound Absorption?

...Make sure you get FIRE and MOISTURE RESISTANCE, too!



REPORTS of fires taking hold on combustible duct linings and causing serious damage come as a warning from the National Board of Fire Underwriters!*

With life and property in danger, the use of a material that provides fire-safety, as well as sound absorption, becomes a necessity.

J-M Air Acoustic Sheets perform exactly that function. This sound-absorption material cannot burn or smolder . . . will never contribute to fire spread. And being moisture resistant, Air Acoustic Sheets are exceptionally free from the dangers of humidity.

Furthermore—their high rate of sound absorption (up to 80%) . . . light weight, assuring easy handling and application . . . the permanence of their mineral composition . . . their high transverse strength and low thermal conductivity . . . all make this material outstanding in its field. For complete data on J-M Air Acoustic Sheets, write Johns-Manville, 22 E. 40th Street, New York City.

*A recent bulletin from the N.B.F.U. reports several cases in which combustible duct linings caught fire and contributed to greatly increased loss.

JM Johns-Manville
AIR ACOUSTIC SHEETS FOR DUCT LINING

Burnhams

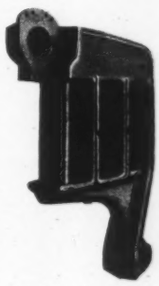
Low Cost Replacement Boiler

FRANKLY now, why spend money tearing out and then repairing the tear-out on a building, in order to replace a boiler? Why do it, when the old boiler can be left where it is, or moved to one side, and a Burnham Cast Iron Twin Sectional put in its place. No tear-outs. No repairs.

None of the Burnham sections are so big they won't slip through any average size door. Still, it is a big boiler having a 50-inch grate.

If you prefer Welded Steel Boilers, Burnham also makes them.

Both are built to a specification. Both are backed by a reputation.



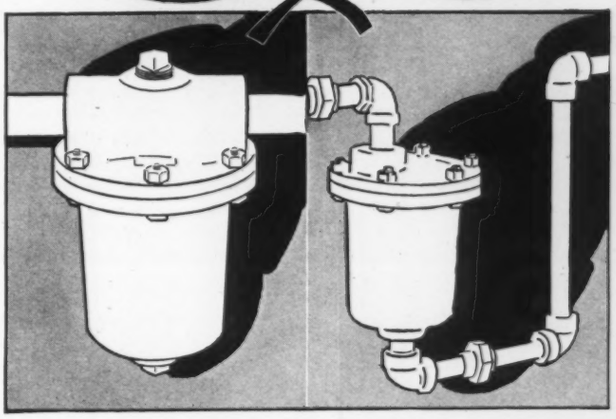
Sections are reversible. There are no rights and lefts. Every right is also a left. Every left is also a right.

Burnham Boiler Corporation

IRVINGTON, NEW YORK
ZANESVILLE, OHIO

Foreign Export Department
116 Broad Street, New York

Why we say the Only COMPLETE STEAM TRAP



For this installation you buy:
1 Anderson Super-Silvertop trap. No fittings—simply install the trap as a part of the pipe.

Labor time:
Fitting 2 threaded joints.

Comment:
No unions needed since trap is inspected or repaired right on the line.

For this installation you buy:
1 old style trap
6 nipples • 4 elbows
2 unions

Labor time:
Fitting 14 threaded joints and closing 2 unions.

Comment:
Unions required since pipe must be taken apart in order to inspect or repair trap.

NO OTHER TRAP has all the advantages (listed below) that go to make Anderson Super-Silvertop a complete steam trap.

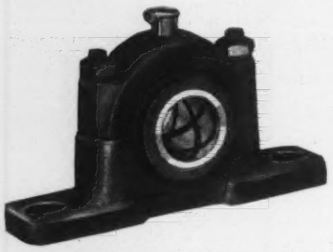
- Connect it either straight-in-line or as an elbow—no extra fittings to buy or install.
- Is inspected, cleaned or repaired without disturbing any pipe connection.
- Needs no strainer—handles ordinary dirt and scale.
- Operates on vacuum as well as pressure, proof that it does not leak steam.
- Averages 76% greater capacity than ordinary traps.
- With snap action Thermal Air Eliminator, it gives instant heat to the steam system.

Anderson Super-Silvertop steam traps are the greatest steam trap value today because they are complete.

THE V. D. ANDERSON CO.
1949 West 96th Street • Cleveland, Ohio

Anderson SUPER-SILVERTOP

Cut-away section of ball of Randall Standard Pillow Block



Randall Standard Pillow Block assembled.
(Patented)

BEARINGS THAT STAY YOUNG IN SERVICE

When most bearings are worn out, Randall Bearings are still giving dependable, QUIET, smooth service.

The Randall Standard Pillow Block, shown here, is the fastest selling item of its type on the market. It is self-lubricating, self-aligning and operates with minimum running torque.

Blowers equipped with Randall Bearings deliver air throughout the system WITHOUT METALLIC BEARING NOISE.

Write for our new catalog, or outline any special bearing problems you have.

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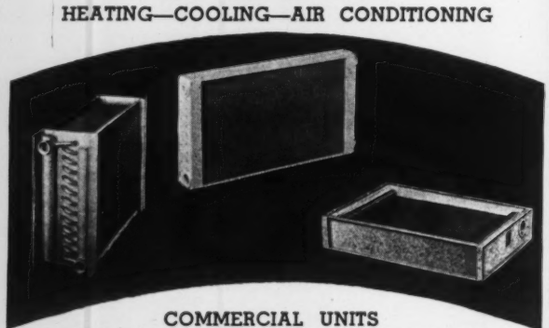
COILS for
HEATING—COOLING—AIR CONDITIONING



COMMERCIAL UNITS
CONTINUOUS TUBE COILS—BLAST UNITS

For years Young heat exchange surfaces have established a standard of quality. Engineers and Heating Contractors can select a unit from the Young line to meet exact specifications with the utmost confidence of its proven performance. Each unit is light weight, compactly designed and properly tested under actual operating conditions to do a definite heating or cooling job. Write for descriptive literature.

OFFICES IN ALL PRINCIPAL CITIES

YOUNG RADIATOR Company
RACINE, WIS.

of them takes a bath and that each bath requires 3 gal of hot water. Then the demand per 10 min will be $28 \times 3 = 84$ gal, or at the rate of 504 gal per hr. This is only a rate of demand and it is difficult to conceive, of course, that so rapid a draw down ever will occur during a full hour.

I would install a storage tank having a capacity of about double the 10 min demand or say 150 gal. Suppose that 2 hr are allowed for heating the contents of the storage tank. Then 75 gal $\times$ 8.3 lb $\times$ 100 deg temperature rise = 62,250 Btu per hr. This reduced to equivalent direct radiation becomes approximately 260 E.D.R. The total steam demand then will be 750 sq ft radiation + 260 E.D.R. + allowance for pipe surface losses, say 20 per cent = 1212 E.D.R.; which may be the basis for determining the size of the steam supply main. The condensation from this volume of steam under maximum demand will be about $1212 \times 0.25 = 303$ lb. No doubt the entering cold domestic water will be at about 40 F. The 303 lb per hr of condensation will be at about 200 F. It is proposed to discharge this clean hot water to the sewer.

Why not save the heat in this condensation? An easy way to accomplish this saving will be by passing the hot water around copper coils in a small preheater tank, with the cold domestic water passing through these coils on its way to the main steam heated domestic water storage tank and with recirculation arrangements for the water. By this means the 303 lb per hr of water could be cooled from 200 F to at least 150 F, and to a temperature much cooler than this during peak demands when large volumes of cold water are entering the system. 303 lb $\times$ 50 deg = 15,150 Btu per hr, or nearly one-quarter of the hourly water heating demand.

This heating plant can operate at very little pressure above atmospheric pressure and if it is well installed without leakage can operate economically for long periods of time at less than atmospheric pressure.

A pressure reducing valve combined with a thermostat in the common room could vary the steam pressure and corresponding temperature in accordance with the demand.

There should be a high grade automatic air escape valve at the outlet end of the return main, arranged to permit escape of air but preventing any return of the air. A single float trap at the outlet end of the return main would allow the condensation to enter the economizer without permitting air to enter. If the steam main makes a single circuit down one side of the building and back along the other, the length will be about 240 ft. It is conservative to allow a pressure drop due to friction through such a system as this, of ⅛ lb. In other words, under maximum demand a pressure gauge at the entrance to the steam main would register about 2 oz higher than a similar gauge at the outlet from the steam main.

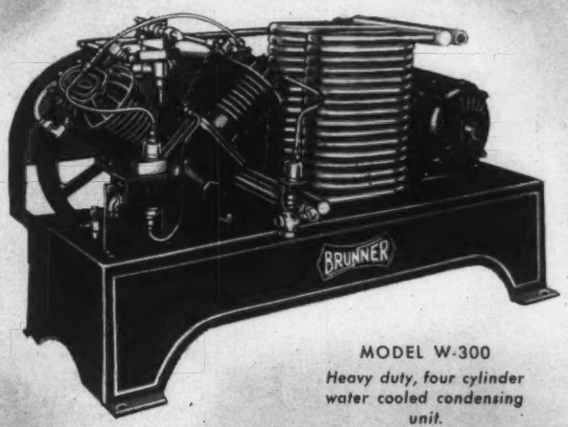
The hot water heating coil preferably will receive steam at 12 lb pressure directly from the main and will have a separate trap. The pressure reducing valve will supply only the direct radiators and will have a separate trap.

Therefore the heating system main of 240 ft length must serve 750 E. D. R. + 20 per cent = 900 E. D. R. at ⅛ lb drop per 100 ft of run. From standard pipe capacity tables this load will require a 3 in. main with 1¼ in. return, though it is a load not greatly in excess of that which would be served by a 2½ in. main.

The basement or space under the joists should be ventilated in some manner, preferably by a chimney leading through warm rooms of the building and exhausting air into the basement out of the common room. Otherwise the warm basement may be damp due to absorption of moisture from the ground.

The occupants of the building are quite likely to open the windows at night, with the radiator valves closed. The bedroom floors then may be chilled below the dewpoint temperature of the basement air, and this moisture will condense against and be absorbed by the floor and the joists, which will rot out with surprising speed. Experience with buildings heated as this build-

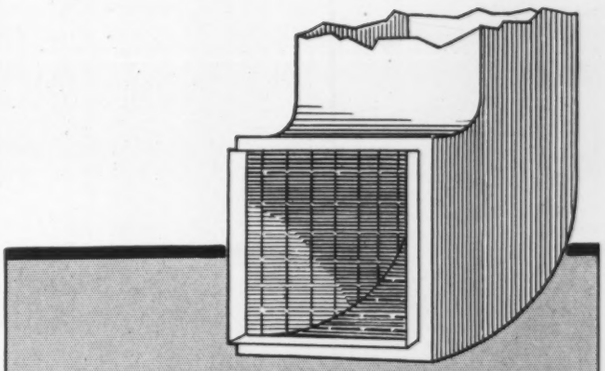
HERE'S WHERE AIR CONDITIONING *Efficiency* BEGINS



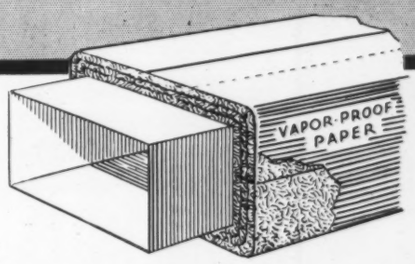
MODEL W-300
Heavy duty, four cylinder water cooled condensing unit.

The Condensing Unit is the "business end" of the air conditioning system . . . the point where operating efficiency begins. Specify BRUNNER for this all-important spot, safe in the knowledge that Brunner efficiency and ruggedness will endure for many years, with low operating cost and minimum repairs . . . A wide range of models for all installations up to 15 tons of refrigeration. Catalog on request. Brunner Manufacturing Company, Utica, N. Y., U. S. A.

BRUNNER
for Greater Dependability!



Ozite HAIR FELT
AIR DUCT LINING AND DUCT INSULATION



You Need Both For Maximum Efficiency

To stop vibrations and noise . . . line air ducts with Ozite Hair Felt Air Duct Lining! Its coefficient of sound absorption is unusually high at the frequencies at which most fan and blower noises occur. Ozite Duct Lining is available ½" thick or 1" thick in bats with paper cover or felted with burlap backing and latex facing.

To keep heat where you want it . . . *outside* in summer and *inside* the air ducts in winter . . . insulate with Ozite Hair Felt Air Duct Insulation! Its thermal conductivity is only 0.246 B.t.u.'s. It is supplied in rolls, 3 or 6 feet wide of any length up to 50 ft. It should be encased, after application, in a water-vapor-proof paper envelope.

Both products are odorless, fire-resistant and repellent to rodents and vermin. Complete specifications and details on request. Write today!

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AEROFIN
Heat Exchange Surface
PROVED PERFORMANCE

FOR HEATING or COOLING
Write for Literature
AEROFIN CORPORATION
NEWARK, N. J.

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RE-COOLING NOZZLES

Monarch Fig. B-8 and new B-8-A cast Brass nozzles, for cooling condensing water in spray ponds, use no internal vanes or deflectors which might facilitate clogging. Available in capacities up to 104 G.P.M. at 10 lbs.

For industrial air washers we recommend our ¼" Fig. 631 (female) or ⅜" Fig. 629 (male pipe) delivering 68 G.P.H. at 20 lbs. pressure.

For air conditioners we recommend our all brass Fig. F-27 (⅛" or ¼" female pipe) or Fig. H-261 (¼" male pipe) nozzles. Produce a very fine spray using city water pressure.

Write for New Catalog 6-A

MONARCH MFG. WORKS, INC.
2728 E. Westmoreland St., Philadelphia, Pa.

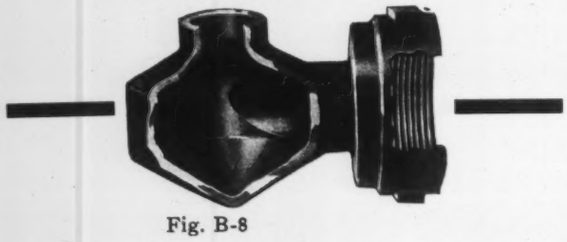


Fig. B-8

ing is to be heated and occupied as it will be occupied, leads m to suggest that the floor be protected by an intermediate laye of insulation between the rough floor and the finished floor an by having a waterproof membrane under it. This might be don by placing light roofing on top of the joists under the floor, o might be done better by placing some inexpensive air tight ceilin below the joists.

If the supply main is installed with a pitch toward its remot end of at least ½ in. in 20 ft 0 in., a 1 in. float trap at the en of the main, say 4 ft beyond the last branch, will drain it. If there are two circuits, each circuit must have a separate trap and there must be no untrapped connection between the supply main and the return main. There must be no untrapped connection between any two circuits of the supply main.

There are 900 E. D. R. to be supplied with low pressure steam and each sq ft condenses about ¼ lb per hr of steam, or there must be delivered through the reducing valve at an initial pressure of 12 lb, 225 lb per hr of steam.

The formula for computing the size of a reducing valve is as follows, provided that the reduced pressure is less than half the initial pressure:

$$\text{Size in sq in.} = \frac{\text{lb per hr of steam} \times \text{initial volume per lb} \times \text{cu in. per cu ft}}{\text{Seconds per hr} \times \text{velocity through valve in ft per sec} \times \text{in. per linear ft}}$$

$$\text{Substituting: } \frac{225 \times 15.2 \times 1728}{3600 \times 250 \times 12} = 0.547 \text{ sq in.}$$

The internal area of a 1 in. pipe is 0.864 sq in. and of a ¾ in. pipe is 0.533.

The formula is exceedingly conservative, and a ¾ in. reducing valve is recommended. The reducing valve should have a pressure gauge at its inlet and one at its outlet. There should be unions and valves at each side of the reducing valve so as to permit its removal for maintenance purposes, with a valved bypass around the assembly so as to permit emergency operation. The reducing valve need not have an expanded outlet. The pipe can be increased from ¾ to 3 in. with fittings close to the valve outlet.

It is wise usually, if there is any danger that the steam pressure might build up in excess of the 12 lb quoted as the initial pressure, to install a small safety valve on the low pressure side having its outlet connected to a whistle, so that attention will be called to the excess pressure. There are many instances in which the pressure and corresponding steam temperature in small plants of this type has increased to such an extent as to expand the thermostatic elements in the traps so much as to ruin them.—SAMUEL R. LEWIS.*

*Consulting Engineer, Chicago, Ill., Member of Board of Consulting and Contributing Editors.

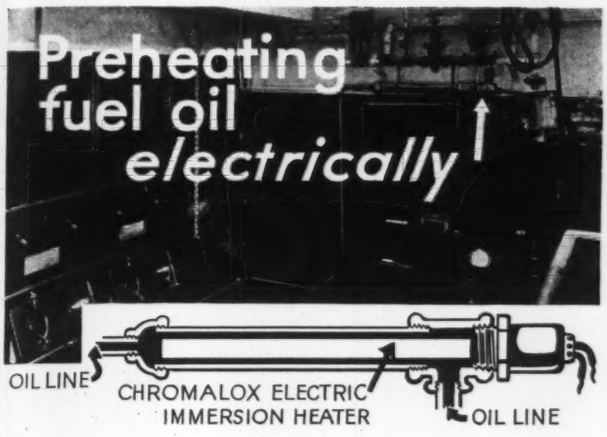
Recent Trade Literature

For your convenience in obtaining copies of bulletins, see coupon on page 156

No. 2306. AIR CONDITIONERS: Airtemp, Inc., Subsidiary of Chrysler Corp., Leo St., Dayton, Ohio. Bulletin announcing "All-in-One" summer air conditioner for stores, shops and offices, requiring floor space of 20 x 32 in. and standing 7½ ft high overall. Capacity is 3 hp.

No. 2307. AIR CONDITIONERS: Davis Engineering Corp., 1064 E. Grand St., Elizabeth, N. J. 4 p. bulletin describing "Chillblast" air conditioner, cooler and dehumidifier, a central type industrial unit consisting of casing containing refrigerating coils cooled by direct expansion or brine over which is maintained a constant flow of water. Information on operation and applications is included.

No. 2308. AIR CONDITIONERS: Davis Engineering Corp., 1064 E. Grand St., Elizabeth, N. J. 4 p. bulletin describing



For starting furnaces burning a heavy grade of fuel oil, preheat the oil electrically, use the simple and inexpensive arrangement illustrated. This is constructed of standard pipe and fittings, making up a housing for a Chromalox electric immersion heater around which the fuel oil flows on its way to the burners. The 200 deg. F. to which the oil must be preheated in order to burn efficiently, is quickly developed by this unit.

This is one of thousands of simple but valuable industrial applications of electric heat, using Chromalox heating units which are available in every form, type and capacity required. The use of these units is invariably characterized by simplicity and compactness, as well as fire safety and operating economy.

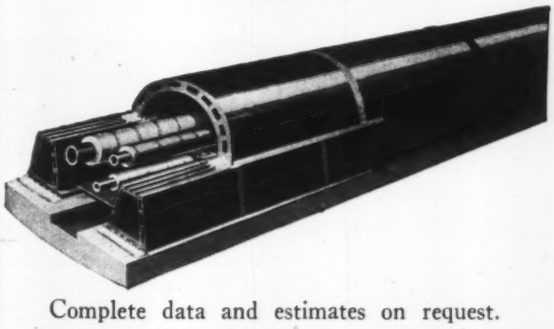
Chromalox units are used to electrically heat air, gases, fluids, machine parts, processes, etc. The experienced Wiegand engineering staff is ready to aid in developing any specific application. For general information, the Chromalox Book of Electric Heat, a valuable treatise on the subject, is mailed upon request.

EDWIN L. WIEGAND COMPANY
7610 Thomas Blvd. Pittsburgh, Pa.

THERM-O-TILE

The Conduit for Underground Steam Lines

Simplest
Strongest
Most Efficient



Complete data and estimates on request.

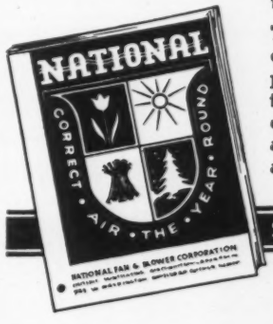
Sold and Installed by Johns-Manville Construction Units in all Principal Cities.

H. W. PORTER & CO., INC.
825 FRELINGHUYSEN AVE., NEWARK, N. J.

SETS THE PACE

● Illustration shows Single Inlet Blower, Arrangement No. 3, used for both supply and exhaust. Their field covers the entire air conditioning industry. Sizes and types to cover all needs.

The National Line is complete including unit heaters, air washers, propellor fans, exhausters, ventilators, furnace blowers, blast coils, etc. We have a 100% agency proposition. Write for available territory.



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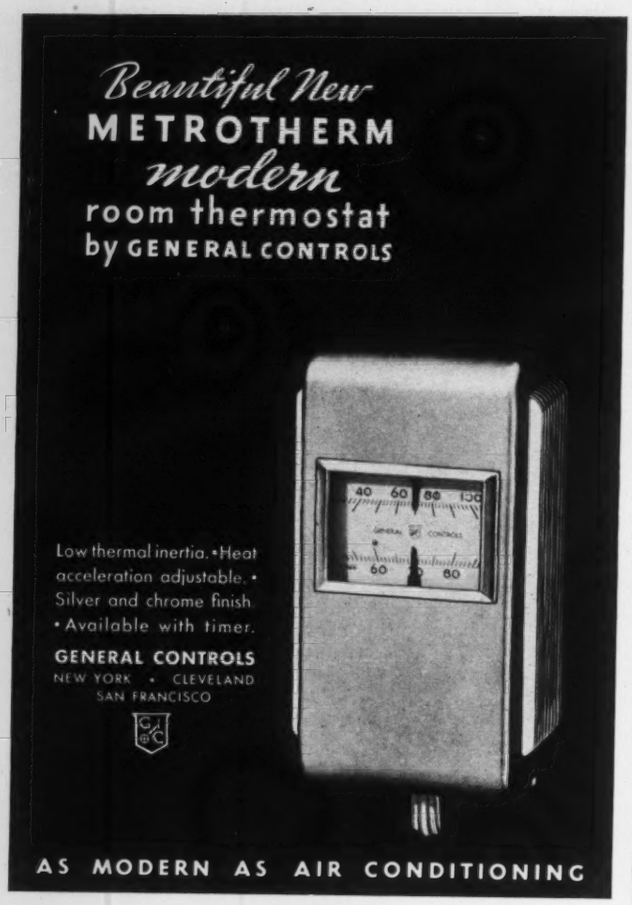

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proof box type air conditioner for bakeries, including list of features and advantages, drawings and tables of sizes and dimensions.

No. 2309. AIR CONDITIONING: Auditorium Conditioning Corp., 17 E. 42nd St., New York, N. Y. 4 p. bulletin describing in detail the "Auditorium" licensing plan and what it means to designers and buyers. How permission to utilize the features of "Auditorium" systems can be secured by an owner, general contractor, sub-contractor, or air conditioning contractor, is explained, and information is given on central station and unitary air conditioning systems.

No. 2310. AIR CONDITIONING UNITS: Fedders Mfg. Co., 57 Tonawanda St., Buffalo, N. Y. 36 p. catalog on "All Season" air conditioning units illustrating and describing new line built in capacities from 3 tons up for use with refrigerant, cold water, steam, or hot water. Useful working data and specifications, as well as descriptive material, numerous charts, rating tables, and examples worked out in detail, are included.

No. 2311. AIR CONDITIONING AND HEATING ACCESSORIES: Borg Warner Service Parts Co., 310 S. Michigan Ave., Chicago, Ill. 180 p. catalog listing parts and accessories for air conditioning and automatic heating units and electrical refrigeration.

No. 2312. AIR FILTERS: Amirton Co., Inc., 49-53 E. 21st St., New York, N. Y. 4 p. bulletin describing "Amirglass" filters of the automatic and manual cleaning types, giving a table of ratings, and describing accessory equipment.

No. 2313. ALLOYS: International Nickel Co., Inc., 67 Wall St., New York, N. Y., 12 p. technical data bulletin on nickel alloy steels in petroleum production equipment, of general interest as the same principles apply to the selection of materials for other types of heavy duty machinery. Detailed descriptive material is presented as well as tables summarizing general requirements and properties of nickel alloy steels.

No. 2314. BEARINGS: Johnson Bronze Co., New Castle, Pa. 32 p. bulletin covering general line of phosphor bronze bushings and bar bronze. Progressive size listings and price lists are included.

No. 2315. BLOWER WHEELS: Janette Mfg. Co., 556-558 W. Monroe St., Chicago, Ill. 4 p. bulletin on blower wheels of various types for oil burners, stokers, gas burners, and air conditioning equipment applications, illustrating and describing features of construction.

No. 2316. COMFORT COOLERS: McQuay, Inc., 1600 Broadway, N. E., Minneapolis, Minn. 8 p. catalog of comfort coolers of suspended propeller type, giving general data on construction features, illustrating features and installations, and presenting tables of cooling and heating capacities using cold water or brine for cooling; hot water or steam for heating; and direct expansion types. A typical estimate for a comfort cooler installation is shown in detail.

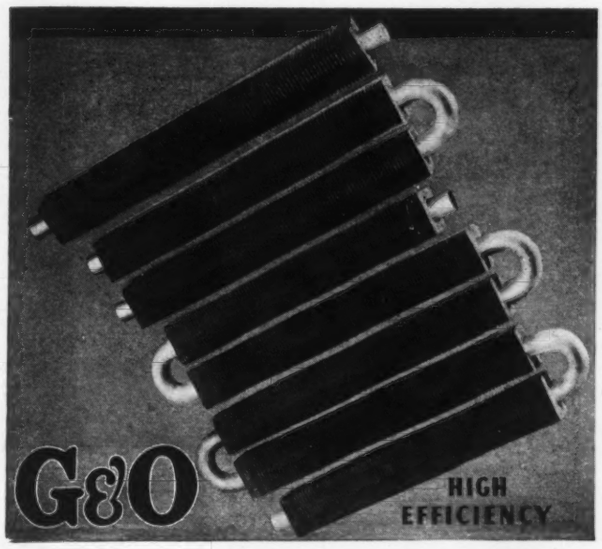
No. 2317. CONDENSATE RETURN UNITS: Cameron Pump Div., Ingersoll-Rand, 11 Broadway, New York, N. Y. 8 p. bulletin on condensate return "Motorpumps," describing and illustrating the various types and giving specifications, dimensions, capacity table, and information on installation.

No. 2318. CONDUIT SYSTEM: H. W. Porter & Co., Inc., 825 Frelinghuysen Ave., Newark, N. J. 10 p. booklet showing step by step installation of the "Therm-O-Tile" steam conduit system.

No. 2319. CONDUITS: Ric-wiL Co., 1562 Union Trust Bldg., Cleveland, Ohio. 4 p. bulletin showing recent interesting installations of underground conduit for steam lines and describing features of this type of conduit.

No. 2320. CONTROLLERS: American Schaeffer & Budenberg Division, Consolidated Ashcroft Hancock Co., Inc., 11 Elias St., Bridgeport, Conn. 8 p. bulletin on "American" wide range, non-indicating controllers for temperature and pressure, illustrating and describing the instruments in detail and listing applications.

No. 2321. CONVECTORS: Commodore Heaters Corp., 11



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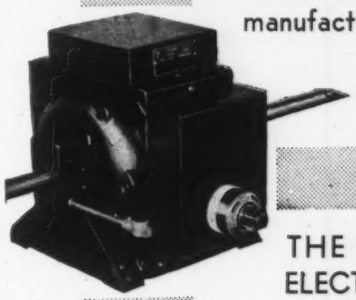

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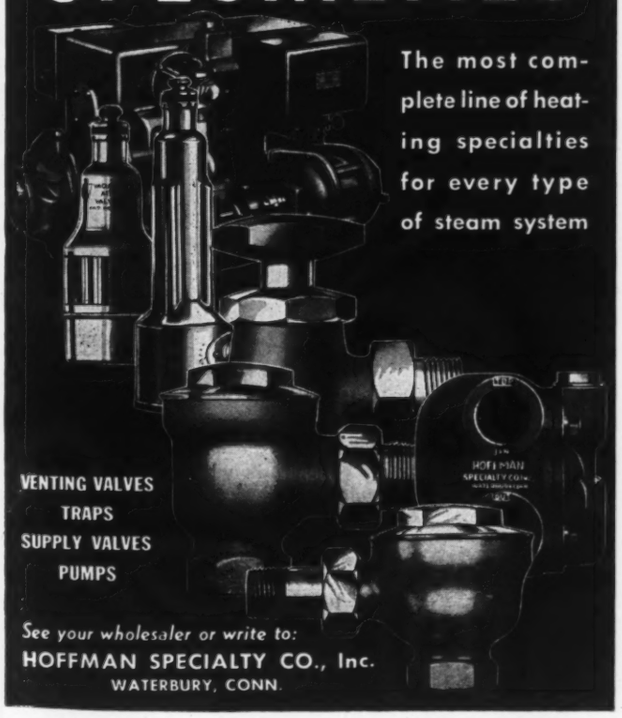

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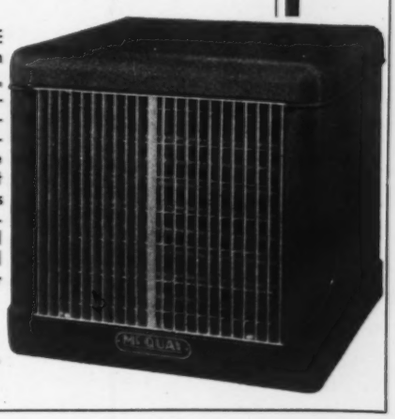

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Installation Data

Total length and width	52'x22'
Front part length	20'
Rear part length	32'
Height of Ceiling	16' 6"
Velocity in Duct and ANEMOSTAT Neck	1430 fpm
Temperature Differential	20 deg.
Total Volume of Air	2500 cfm
1 ANEMOSTAT distributes	1500 cfm
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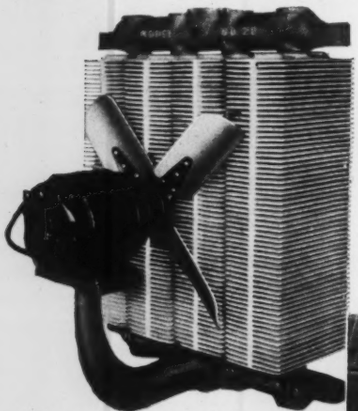

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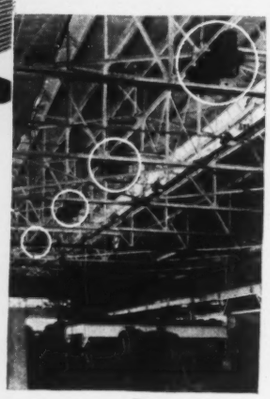
Before you decide on any heating equipment.



These heating sections are all-cast aluminum. In them there are no soldered joints—no pressed joints—no rolled joints—no unions. It's a sturdy construction that cannot bend out of shape and will not break. Neither can these sections be affected by electrolysis. That's why in 8 heating seasons there have been no failures due to leaks or breakdowns. Once installed Grid Unit Heaters become permanent equipment to last indefinitely.

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No. 2322. CYCLE COUNTERS: Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. 4 p. bulletin describing cycle counter for indicating time of operation of any apparatus which can be arranged to open or close an a-c circuit of known frequency. Applications, construction and operation are described.

No. 2323. FLOW METERS: Brown Instrument Co., Div. of Minneapolis-Honeywell Regulator Co., 4534 Wayne Ave., Philadelphia, Pa. 56 p. catalog on electrical and mechanical flow meters of indicating, recording and integrating types. Descriptions of installations for various services are included. How flow meters operate is explained and detailed data are given on the various types for different applications.

No. 2324. FLUID METERS: Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland, Ohio. 40 p. catalog of fluid meters for steam, liquids and gases, indicating how any desired combination of indicating, recording and integrating features may be combined with a flow mechanism for steam, liquids or gases under high, low, or medium pressures. The publication is divided into 10 sections including data on the equipment and its installation.

No. 2325. HEATING EQUIPMENT: Burnham Boiler Corp., Main St., Irvington, N. Y. 28 p. catalog of heating equipment including cast iron boilers for coal, oil and gas; steel boilers for all fuels; hot water supply boilers; slenderized and tube radiators; and specialties, including unit air conditioner, air valves, flexible headers, and unit heaters. Features of the various items of equipment are described and illustrated in detail and rating and dimension tables are given.

No. 2326. LIQUID LEVEL CONTROLLERS: Foxboro Co., 106 Neponset Ave., Foxboro, Mass. 24 p. bulletin discussing "Stabilog" controllers for liquid level and application methods for automatic control of liquid level in continuous flow processes, Of especial interest is a 2 p. chart and discussion of flow problems and their solution by means of these controllers.

No. 2327. LUBRICATION: Texas Co., 135 E. 42nd St., New York, N. Y. Issue of "Lubrication," Vol. 23, No. 2, devoted to heat exchange as it relates to effective lubrication, and discussing among other subjects, steam cooling, lubricating heat exchangers, pump circulation, etc.

No. 2328. MOTORS: Fairbanks, Morse & Co., 900 S. Wabash Ave., Chicago, Ill. 4 p. bulletin describing line of polyphase wound rotor or slip ring ball bearing induction motors. constant speed or adjustable varying speed. The high starting torque and low starting current characteristics make them suited to applications where the relatively high starting current of squirrel cage motors would be objectionable.

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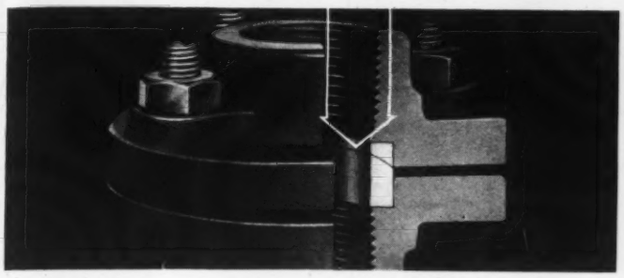

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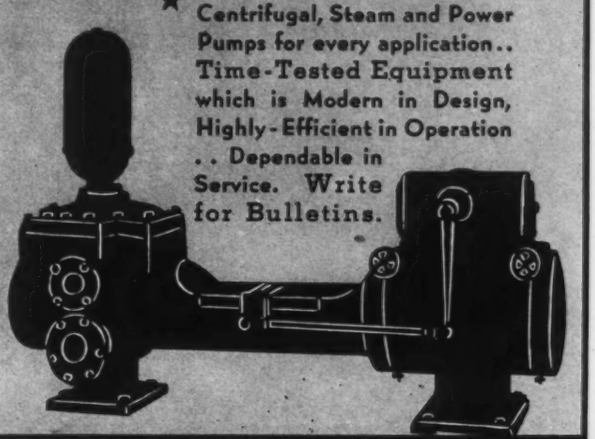

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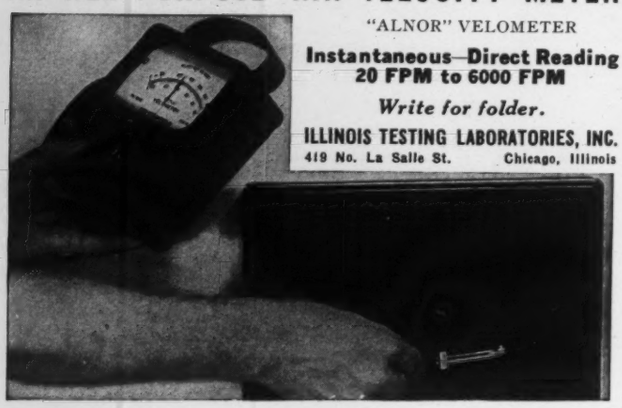

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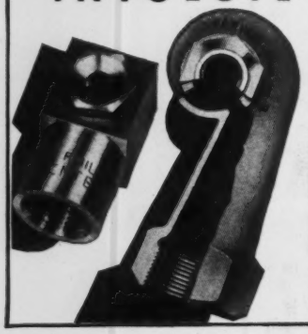

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No. 2330. SHEET METAL: Republic Steel Corp., Republic Bldg., Cleveland, Ohio. 24 p. booklet, "The Path to Sheet Metal Permanence," containing over 60 photographs showing uses for sheet metal and giving information on rust resistance, forming and welding properties, physical properties, and constants.

No. 2331. STOKERS: Detroit Stoker Co., General Motors Bldg., Detroit, Mich. 24 p. catalog devoted to the use of this manufacturer's stokers in prominent educational institutions, including private and parochial schools, colleges, universities and public schools in the U. S. and Canada. Features of the stokers are described and information is given on savings achieved in a number of cases.

No. 2332. SWITCHES: General Electric Co., 1 River Road, Schenectady, N. Y. 8 p. bulletin describing rotating cam switches designed especially for built-in control applications and adaptable to a variety of electrical functions and machine requirements.

No. 2333. TEMPERATURE AND PRESSURE CONTROLLERS: C. J. Tagliabue Mfg. Co., 540 Park Ave., Brooklyn, N. Y. 32 p. catalog on "Tag" non-indicating temperature, pressure, and time controllers, giving in addition to instrument listings, a comprehensive description of the different types of controllers, their applications, and how they work.

No. 2334. UNIT HEATERS: Airecon Industries, Inc., 2648-2653 Botsford Ave., Detroit, Mich. 4 p. bulletin on "Acofin" suspended type unit heaters, describing their features in detail and giving capacity and engineering data.

No. 2335. UNIT HEATERS: Airtherm Mfg. Co., 1474 S. Vandeventer Ave., St. Louis, Mo. 16 p. catalog covering "Airheator" line of centrifugal fan unit heaters giving specifications, illustrating the various assemblies, and including a series of capacity tables. Data on piping and temperature control are included.

No. 2336. WATER METERS: Worthington-Gamon Meter Co., Harrison, N. J. 8 p. bulletin on "Watch Dog" heavy duty type disc water meters, describing construction and giving capacities, dimensions, and weights.

No. 2337. WASTE HEAT BOILERS: Davis Engineering Corp., 1064 E. Grand St., Elizabeth, N. J. 12 p. bulletin on "Paracoil" waste heat boilers, discussing conversion of waste heat for usable service, how to estimate how much heat can be converted, recovering heat from jacket water, and describing the equipment in some detail.

No. 2338. WELDING ELECTRODES: Page Steel & Wire Div., American Chain & Cable Co., Inc., 929 Connecticut Ave., Bridgeport, Conn. 16 p. booklet on "Hi-Tensile F" welding electrodes, describing the shielded arc electrode and method of welding and illustrating the procedure for various types of welds.

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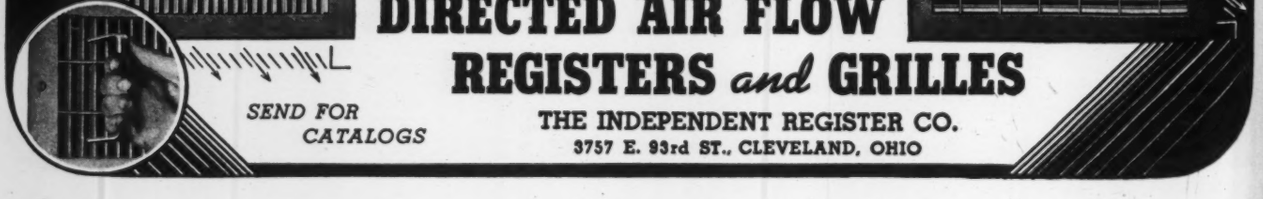

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